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DEVELOPMENT OF AN ANTI-JACKKNIFING SYSTEM FOR TRACTOR SEMITRAILERS: FEASIBILITY STUDY; PHASE 1, STAGE 2

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By
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T E S Limited

May 1990

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The contents of this report reflect the views of the contractor and not necessarily the official views or policies of the Transportation Development Centre of Transport Canada.

Un sommaire en français de ce rapport est inclus avant la table des matierès.

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16. Abstract

This study investigated the feasibility of anti-jackknife systems used to control tractor jackknife in tractor-semitrailers. It was a follow-up to an earlier study for which the results have already been reported. Computer simulation was used as the main analytical tool to determine the performance characteristics of both active and passive anti-jackknife devices. The study also looked at the effects on the resulting jackknife characteristics of varying the braking distribution and the surface friction values.

The work showed that proper distribution of available braking forces significantly reduced the severity of jackknife. The importance of brake proportioning based on normal loading was also reinforced. With respect to varying friction surfaces, it was discovered that it is the difference between the peak and sliding friction coefficients that determine the severity of the resulting jackknife and not the magnitude of the coefficients.

All of the anti-jackknife devices simulated showed merit in controlling tractor jackknife, but two in particular (the active device and the passive variable damping device) performed significantly well.

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16. Résumé

La recherche a porté sur la faisabilité de divers dispositifs permettant de contrecarrer les mises en charnière des véhicules articulés (ensembles tracteur/semi-remorque). Elle complétait une recherche antérieure dont il a déjà été rendu compte. La simulation sur ordinateur a été le principal outil utilisé pour analyser les caractéristiques des dispositifs tant actifs que passifs étudiés et déterminer leur efficacité en fonction de répartitions de l'effort de freinage et de coefficient de frottement variés.

La recherche a montré qu'une répartition judicieuse de l'effort de freinage réduit considérablement l'intensité des mises en charnière et qu'un dosage de l'effort de freinage en fonction des réactions pneus-chaussée est vital. Pour ce qui est de l'influence des coefficients de frottement variables, il a été constaté que l'écart entre les valeurs maximale et minimale de ce coefficient a plus d'influence sur les mises en charnière que ces valeurs elles-mêmes.

Tous les dispositifs antimise en charnière étudiés ont réussi plus ou moins à contrecarrer ce phénomène, mais les plus efficaces ont été le dispositif actif et le dispositif passif à amortissement variable.

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Systèmes antimise en charnière, dispositifs antimise en charnière, mise en charnière, ensembles tracteur/semi-remorque, simulation assistée par ordinateur, véhicules articulés	Le Centre de développement des transports dispose d'un nombre limité d'exemplaires.

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SUMMARY

In late December 1989, T E S Limited was contracted by the Transportation Development Centre to continue work from an earlier project (Stage 1) relating to the feasibility of anti-jackknife devices for tractor semitrailers. The purpose of this study (Stage 2) was to examine in more detail the characteristics of tractor jackknife and potential methods of controlling it through devices located about the fifth wheel. In particular, the following areas were investigated:

a) detection of the onset of jackknife,
b) distinguishing jackknife from both trailer swing and the transient responses of normal manoeuvres,
c) investigation into the characteristics of jackknife induced through varying brake distributions,
d) investigation into the effects varying friction surfaces had on jackknife, and
e) evaluation of the performance, and determining the merits of both passive and active type anti-jackknife devices.

The investigation focused primarily on results obtained through computer simulation using the "HSRI/MTC Simulation Program for a Tractor-Trailer in the Horizontal Plane".

Our earlier study, Stage 1, (1) indicated that jackknife is a serious problem associated with tractor semitrailers. Research showed that approximately 8% of all combination unit accidents occur because of jackknife. To examine the problem, two tractors (short and long wheelbase) and two semitrailers (short and long wheelbase) were chosen such that when combined would produce four combinations representative of a wide range of tractor semitrailers on the road today. Computer simulation was used as the main analytical tool in studying the characteristics of jackknife and the effectiveness of the anti-jackknife devices in controlling vehicle instability.

Both forms of jackknife instability, tractor jackknife and trailer swing, were examined in Stage 1. It was reported that jackknife instability occurs because of the saturation (or limitation) of lateral resistive forces at the tire road interface. It was also observed that tractor jackknife and trailer swing behave in dissimilar manners. The articulation angle in jackknife diverges more quickly than it does in trailer swing, and in an opposite direction. Trailer swing also proved to be a milder form of instability as it tends to regain stability without any form of compensation.

A number of jackknife control devices were studied in Stage 1 to determine their effects on both tractor jackknife and trailer swing. The devices were categorized as locking, damping, limiting, or a combination thereof. Locking consisted of preventing any further articulation at the onset of jackknife or trailer swing. Damping, which can take many forms, consisted of slowing down the articulation, thereby providing the driver with more time to react. Limiting consisted of restricting the articulation range during highway travel. All devices studied showed merit in controlling tractor jackknife, but demonstrated little or no ability to control trailer swing. In fact,

some devices presented a situation worse than not having an anti-jackknife device present. This adverse situation was due directly to the large torques produced during trailer swing. When trailer swing was inhibited by a restraining torque, the rear tractor tires became saturated and the entire vehicle rotated out of control.

The results of our earlier study indicated that further efforts were still required to examine some areas of the jackknife phenomenon in more detail. Specifically, it was determined that research efforts should be primarily concentrated at attempting to control tractor jackknife only.

The scope of the Stage 2 project was to investigate the feasibility of anti-jackknife devices that control tractor jackknife by restraining articulation. Thus, devices such as anti-lock brake systems (ABS), load sensing with brake proportioning, and anti-slip control systems were not investigated. To carry out the project, two of the vehicle configurations used in Stage 1 were also used in Stage 2. Namely, a long wheelbase tractor (5.6 m) combined with a long wheelbase semitrailer (14.6 m long) labelled in Stage 1 as configuration 1, and a short wheelbase tractor (3.6 m) combined with a short wheelbase semitrailer (8.2 m long) labelled as configuration 4, were used. The two configurations were simulated on two manoeuvres, the constant radius turn and the lane change, at various speeds that equated to theoretical lateral accelerations of 0.1 g, 0.2 g and 0.3 g on the constant radius turn.

The approach taken toward achieving the project objectives was to first simulate the configurations traversing the two manoeuvres at the given speeds in an unbraked situation to provide ideal results against which later simulation runs could be compared. The next step was to induce jackknife (and trailer swing) in each instance to provide a worst case situation against which all subsequent simulation runs could be compared. The two sets of runs not only provided a 'performance envelope' which specified a range within which the anti-jackknife devices should fall, but also provided the data required for investigation into jackknife detection.

All simulations in Stage 1 of the project involved jackknife being induced through lock-up of the wheels on the rear tractor tandems. Although this represents a worst case situation, it is relatively infrequent in reality. Five new braking distributions were therefore studied to examine the impact variations in braking had on jackknife. This direction led to investigation into the effects variations in surface friction had on jackknife characteristics. Although it was beyond the scope of the project, this endeavour was pursued because of its importance.

With all the previously mentioned areas researched, a comprehensive investigation into two types of anti-jackknife devices, active and passive, was carried out. A single active device that attempted to redistribute to the semitrailer the increment of angular momentum caused by jackknife was modeled. Five different types of passive devices were also modeled: the spring type device, the linear damping device, the spring/dashpot system, constant damping device, and the variable damping device.

The results obtained from investigation into the effects of various braking distributions showed milder forms of jackknife instabilities than jackknife induced through rear tractor wheel lock-up, as was expected. The investigation also showed that jackknife can be controlled to a degree with proper brake proportioning. In fact, it may be possible to eliminate jackknife, but the entire vehicle would continue, in a skid situation, in the general direction of travel.

It was discovered that the variance between the peak and sliding friction coefficients (μ_p and μ_s) have a greater impact on the degree of jackknife than the magnitude of the friction values. Thus, a friction surface such as wet pavement, with peak and sliding friction values of 0.6 and 0.35 respectively, would result in a more aggressive jackknife than snow, with peak and sliding friction values of 0.26 and 0.25, if jackknife action were initiated. The reason for this difference in severity is because of the sudden drop from a maximum restraining force to minimum when wheel lock-up occurs.

Although a means of detecting the onset of jackknife was not found, an active anti-jackknife device was modeled that used a significant decrease in lateral forces at the tire-road interface as a triggering mechanism. This device attempted to redistribute to the semitrailers the increment of tractor angular momentum associated with the incipient jackknife. The device performed very well by greatly controlling vehicle articulation and maintaining vehicle trajectory. However, as the device suddenly became active once jackknife had begun, its effect on vehicle stability was questionable.

Of the five passive devices, only two, the constant damping device and the variable damping device, showed significant improvement over the unrestrained jackknife situation (all devices did show some improvement, however). The constant torque device eliminated jackknife in one instance; unfortunately, vehicle controllability was significantly hampered when normal driving manoeuvres were simulated because the restraining moment was continuously applied from the moment simulation began. The variable damping device appeared to out perform all the other devices. It managed to control the path of the configuration as effectively as the active type anti-jackknife device, but in a more stable manner. Although articulation was not restrained as well as it was with the constant damping device, the variable damping device allowed the vehicle to track closer to the intended path. It also showed further merit over the constant damping device by providing much greater vehicle control to the driver during normal manoeuvres. In the 1.5 second time window from the point of wheel lock (a conservative night time driver reaction time) the variable damping device held vehicle articulation to less than 16 degrees, which was speculated to allow a driver a good chance to regain control of the vehicle.

Sommaire

Fin décembre 1989, le Centre de développement des transports de Transports Canada a donné à T E S Limited le mandat de poursuivre les travaux d'une phase précédente (phase 1) concernant l'étude de faisabilité sur divers dispositifs antimise en charnière des véhicules articulés (ensembles tracteur/semi-remorque). La phase actuelle, phase 2, visait à approfondir les caractéristiques de ce phénomène ainsi que les méthodes permettant de le contrecarrer grâce à des dispositifs agissant sur l'articulation, et plus particulièrement :

a) de détecter l'imminence d'une mise en charnière;
b) de distinguer entre une mise en charnière, d'une part, et les oscillations de la semi-remorque et la réponse transitoire à des manoeuvres normales, d'autre part;
c) de caractériser les mises en charnières induites en fonction de diverses répartitions de l'effort de freinage;
d) de caractériser les mises en charnières en fonction du coefficient de frottement des chaussées; et
e) d'évaluer le comportement des dispositifs antimise en charnière tant passifs qu'actifs avant de déterminer les avantages et les inconvénients de chacun.

La recherche s'est concentrée sur les résultats des simulations effectuées grâce au programme informatique "HSRI/MIC Simulation Program for a Tractor-Trailer in the Horizontal Plane".

La recherche antérieure avait montré que les mises en charnière sont un phénomène dangereux qui touche couramment les ensembles tracteur/semi-remorque: elles comptent pour 8 p. 100 de tous les accidents arrivés à ce type de véhicules. Pour approfondir la question, on a choisi deux tracteurs (empattements long et court) et deux semi-remorques (empattements long et court), permettant d'obtenir quatre combinaisons possibles représentatives de celles que l'on voit circuler le plus souvent. La simulation sur ordinateur a été le principal outil utilisé pour analyser les caractéristiques des mises en charnière et évaluer l'efficacité des dispositifs permettant de les prévenir.

La recherche antérieure s'était penchée sur les deux formes d'instabilité latérale : mise en charnière du tracteur et oscillations de la semi-remorque. Il a pu ainsi être déterminé que ces instabilités surviennent lorsque les forces latérales que les pneus peuvent transmettre au sol sont annulées, voire dépassées, par les forces latérales induites aux points de contact pneus-chaussée. Il a été également constaté que ces deux formes d'instabilité latérale ne se ressemblent pas, leur effet sur l'angle d'articulation étant plus rapide dans le cas des mises en charnière que dans celui des oscillations, qu'elles agissent dans des sens opposés et que ces dernières ne sont qu'une forme atténuée d'instabilité dynamique, étant donné qu'elles ont tendance à se résorber sans compensation aucune.

La recherche antérieure avait évalué l'effet d'un certain nombre de dispositifs antimise en charnière et anti-oscillations. Ils ont été caractérisés selon l'effet qu'ils produisent : blocage, amortissement, bridage ou une combinaison de ces trois effets. Le blocage inhibe tout mouvement au-delà de l'angle amorçant un mouvement latéral induit. L'amortissement qui peut prendre plusieurs formes, ralentit les mouvements induits, permettant ainsi au conducteur de réagir à temps. Le bridage contient les mouvements induits durant les déplacements autoroutiers à l'intérieur d'une plage donnée. Si tous ces dispositifs possèdent des qualités dans la prévention des mises en charnière, ils ont été peu ou pas efficaces à contrer les oscillations des semi-remorques. En fait, quelques-uns ont eu un effet nettement négatif. Cela s'explique par le fait que, lorsque l'articulation est rendue rigide, les valeurs élevées de moment de rotation induites par les oscillations font perdre aux roues des essieux moteurs leur capacité de guidage latéral, entraînant une tête-à-queue de tout le véhicule.

L'étude antérieure avait montré la nécessité de plus amples recherches sur certains aspects particuliers du phénomène de mise en charnière et l'utilité de s'attaquer à la prévention des mises en charnière seulement.

Les travaux de la phase 2 ont consisté à étudier l'utilité des dispositifs antimise en charnière qui agissent sur l'articulation. Ils ne concernent donc ni les systèmes antiblocage qui dosent l'effort de freinage en fonction des réactions pneus-chaussée, ni les systèmes antipatinage. Deux des quatre configurations utilisées dans la recherche antérieure ont été réutilisées : un tracteur à empattement long (5,6 m) avec une semi-remorque à empattement long (14,6 m), désignée configuration 1 en phase 1, et un tracteur à empattement court (3,6 m) avec une semi-remorque à empattement court (8,2 m), soit la configuration 4 en phase 1. Les simulations faites ont représenté ces deux configurations effectuant deux manoeuvres : virages à rayon de braquage constant et changements de file, à des vitesses variées qui, dans le cas des virages à rayon constant, ont donné l'équivalent de 0,1, 0,2 et 0,3 g.

La méthode mise en oeuvre a consisté d'abord à simuler le comportement de ces deux configurations effectuant les manoeuvres retenues, aux vitesses choisies mais sans freinage, permettant ainsi d'obtenir des courbes "idéales" pouvant servir de base de comparaison pour les simulations ultérieures. L'étape suivante a consisté à induire des mouvements latéraux représentant dans chaque cas les situations les plus défavorables, afin de se donner une deuxième base de comparaison. Ensemble, ces simulations ont permis de dégager une "enveloppe de performances", à l'intérieur de laquelle devraient se situer celles des dispositifs acceptables, et ont fourni les données utiles à la recherche sur la détection de l'imminence d'une mise en charnière.

Les simulations faites dans le cadre de la phase 1 avaient toutes représenté des mises en charnière induites par le blocage des roues des essieux moteurs; bien qu'il représente une situation défavorable, ce phénomène arrive plutôt rarement. Afin de pouvoir mieux étudier le phénomène en fonction de la variation des forces de freinage, cinq nouvelles répartitions de l'effort de freinage ont été choisies. Cette recherche a débouché sur l'idée de caractériser les mises en charnière en fonction du coefficient de frottement des chaussées. Bien qu'il n'entrait pas dans le mandat original, cet aspect de la recherche a été approfondi en raison de son importance.

Une fois toutes ces recherches achevées, une nouvelle a été entreprise sur deux types de dispositifs antimise en charnière, décrits comme actifs et passifs. Un seul dispositif actif a été modélisé, celui qui transfère à la semi-remorque le surcroît de la vitesse angulaire provoquée par la mise en charnière. Par contre, cinq dispositifs passifs ont été modélisés : à ressort, à amortissement linéaire, à ressort et à amortisseur, à amortissement invariable et enfin à amortissement variable.

La recherche sur l'effet des diverses répartitions de l'effort de freinage essayées a mis en évidence des formes d'instabilité latérale atténuées par rapport à celles induites par le blocage des roues, ce qui était prévu. La recherche a également montré que les mises en charnière peuvent être contrecarrées dans une certaine mesure par un dosage adéquat de l'effort de freinage. En fait, il a été montré qu'il est possible de contrecarrer la mise en charnière mais que, en cas de dérapage, le véhicule ira tout droit sur sa trajectoire.

La recherche a révélé que l'écart entre les valeurs maximale et minimale du coefficient de frottement ($_{p}$ et $_{}$) a plus d'influence sur l'amplitude de la mise en charnière que ces valeurs elles-mêmes. En effet, sur une surface mouillée où ces coefficients sont de 0,6 et 0,35, respectivement, la mise en charnière qui en résulterait serait plus dangereuse que sur une surface enneigée, où les valeurs correspondantes sont de 0,26 et 0,25. Cela s'explique par la chute brutale dans la capacité de guidage latéral des pneus en cas de blocage des roues.

Bien qu'aucun moyen permettant de détecter l'imminence d'une mise en charnière n'ait pu être trouvé, on a modélisé un dispositif actif qui se déclenche dès que les forces latérales que les pneus transmettent au sol subissent une chute importante. Ce dispositif transfère aux semi-remorques le surcroît de la vitesse angulaire provoquée par la mise en charnière. Il s'est révélé très utile en réduisant la vitesse angulaire à l'articulation et en conservant au véhicule sa trajectoire. L'inconvénient a été que, une fois le dispositif déclenché, l'effet produit sur la stabilité du véhicule risque d'être nuisible.

Des cinq dispositifs passifs modélisés, deux seulement, ceux à amortissement invariable et à amortissement variable, ont donné des résultats bien meilleurs qu'en l'absence de moyens antimises en charnière (bien que tous les dispositifs aient amélioré la situation. Le dispositif à couple invariable a empêché une mise en charnière dans un cas particulier, mais il gênait grandement la pilotabilité du véhicule étant donné qu'il intervenait avec la même force durant les manoeuvres dites normales. Le dispositif à amortissement variable semble avoir donné les meilleurs résultats, réussissant à conserver au véhicule sa trajectoire avec autant d'efficacité que le dispositif actif, mais avec plus de stabilité. Et bien qu'il n'ait pas réussi à brider l'articulation aussi bien que le dispositif à amortissement invariable, il a quand même conservé au véhicule une trajectoire se rapprochant davantage de la trajectoire voulue. De plus, il a surclassé ce dernier en laissant au conducteur une meilleure pilotabilité durant les manoeuvres normales. Durant

la plage de 1,5 seconde à partir du moment où les roues se bloquent (valeur choisie empiriquement comme représentative du temps de réaction type en conduite de nuit), le dispositif à amortissement variable a maintenu l'articulation à un angle inférieur à 16 degrés, angle que l'on croit être celui qui permet au conducteur de reprendre la maîtrise du volant.

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1.0 INTRODUCTION

This report presents the findings of an indepth investigation into the feasibility of anti-jackknifing systems as they relate specifically to the problem of tractor jackknife. The project, Stage 2, began in December of 1989 to look more closely at the problem of tractor jackknife and how it may be controlled by anti-jackknife devices which act about the fifth wheel of tractor semitrailers.

Our earlier study, Stage 1, (1) indicated that jackknife is a serious problem associated with tractor semitrailers. Research indicated that approximately 8% of all combination unit accidents occur because of jackknife. To examine the problem, two tractors (short and long wheelbase) and two semitrailers (short and long wheelbase) were chosen such that when combined would produce four combinations representative of a wide range of tractor semitrailers on the road today. Computer simulation was used as the main analytical tool in studying the characteristics of jackknife and the effectiveness of the anti-jackknife devices in controlling vehicle instability.

Both forms of jackknife instability, tractor jackknife and trailer swing, were examined in Stage 1. It was reported that jackknife instability occurs because of the saturation (or limitation) of lateral resistive forces at the tire road interface. It was also observed that tractor jackknife and trailer swing behave in dissimilar manners. The articulation angle in jackknife diverges more quickly than it does in trailer swing, and in an opposite direction. Trailer swing also proved to be a milder form of instability as it tends to regain stability without any form of compensation.

A number of jackknife control devices were studied in Stage 1 to determine their effects on both tractor jackknife and trailer swing. The devices were categorized as locking, damping, limiting, or a combination thereof. Locking consisted of preventing any further articulation at the onset of jackknife or trailer swing. Damping, which can take many forms, consisted of slowing down the articulation, thereby providing the driver with more time to react. Limiting consisted of restricting the articulation range during highway travel. All devices studied showed merit in controlling tractor jackknife, but demonstrated little or no ability to control trailer swing. In fact, some devices presented a situation worse than not having an anti-jackknife device present. This adverse situation was due directly to the large torques produced during trailer swing. When trailer swing was inhibited by a restraining torque, the rear tractor tires became saturated and the entire vehicle rotated out of control.

The results of our earlier study indicated that further efforts were still required to examine some areas of the jackknife phenomenon in more detail. Specifically, it was determined that research efforts should be primarily concentrated at attempting to control tractor jackknife only.

The scope of this project, Stage 2, can best be summed up through the following objectives:

- To improve safety in heavy vehicles.
- To determine the feasibility of anti-jackknife devices.
- To detect the onset of jackknife under all circumstances.
- To investigate jackknife characteristics as braking distributions vary.
- To examine passive and active anti-jackknife devices in detail.

No 'sure fire' method of detecting the onset of jackknife (or trailer swing) was found during our Stage 1 investigation. The problem proved to be complex, as it is related to may variables which are mainly configuration dependent. Our approach in this study was to investigate the detection of jackknife in more detail and attempt to distinguish it from both trailer swing and the transient responses of normal manoeuvres.

The influence various brake distributions had on the characteristics of jackknife was another area investigated. Previously, only braking to the point of tire lock-up was employed. Although this represented a 'worst case' situation, questions still arose as to how the performance of a particular anti-jackknife system might react to a milder form of jackknife. For this reason a few 'realistic' braking situations were examined.

In the most general of terms jackknife control devices can be classified as either "active" or "passive". Active anti-jackknife devices become 'active' only when jackknife has been detected. Passive devices are constantly applying some form of controlling torque. Both forms of jackknife control were looked at in our previous study and were examined in more detail during this investigation.

2.0 STAGE 2 OVERVIEW

The purpose of this study (Stage 2) was to examine in more detail the characteristics of tractor jackknife and potential methods of controlling it through devices located about the fifth wheel. In particular, the following areas were investigated:

a) detection of the onset of jackknife,
b) distinguishing jackknife from both trailer swing and the transient responses of normal manoeuvres,
c) investigation into the characteristics of jackknife induced through varying brake distributions,
d) investigation into the effects varying friction surfaces had on jackknife, and
e) evaluation of the performance, and determining the merits of both passive and active type anti-jackknife devices.

The investigation focused primarily on results obtained through computer simulation using the "HSRI/MTC Simulation Program for a Tractor-Trailer in the Horizontal plane". Details on this simulation program can be found in our Stage 1 report (1) *.

2.1 Vehicle Configurations

Two of the four vehicle configurations used in stage 1 were analyzed in this study since they covered a large spectrum of possible configurations and a corresponding variance in response rates. These are:

- Configuration 1: 5.6 m wheelbase tractor/ 14.6 m trailer, and
- Configuration 4: 3.6 m wheelbase tractor/ 8.2 m trailer.

The physical data for the tractors and semitrailers are presented in Tables 1, 2 and 3.

The configuration numbers, 1 and 4, originated with our Stage 1 work and are maintained here for consistency. Configuration 1, the long wheelbase tractor combined with the long wheelbase semitrailer, was chosen because of the large forces which would exist during a jackknife situation. Configuration 4, the short wheelbase tractor combined with the short wheelbase semitrailer, would require a significant reduction in restraining torque during jackknife, but due to its comparatively small inertial values would display much quicker articulation rates in an unstable jackknife situation. Thus, an anti-jackknife device that could respond quickly and effectively to an impending jackknife could be developed and tested.

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* Numbers in brackets designate references listed at the end of the report.

Table 1: Tractor Data (3 Axle)

Parameter	Tractor 1	Tractor 2
Wheelbase (m)	5.6	3.6
Weight (kg)	9,047	6,695
Yaw Moment of Inertia (kg/m^2)	62,000	19,978
Dist. Between Tandem Axles (m)	1.52	1.52
Dist. from CG to Front Axle (m)	2.565	1.666
Dist. from CG to Rear Axle (m)	3.025	1.966
Dist. from CG to Ground (m)	1.118	1.118
Dist. from Rear Axle to Fifth Wheel (m)	0.1524	0.1524
Height of Fifth Wheel Above Ground (m)	1.32	1.32
Half Lateral Dist. Between Centres of Front Tire Contact (m)	1.029	1.029
Half Lateral Dist. Between Centres of Rear Tire Contact (m)	0.991	0.991
Portion of Total Later Load Transfer on Front Axle	0.110	0.110
Tandem Axle Load Transfer Coef.	0.15	0.15

Table 2: Semitrailer Data (2 Axle)

Parameter	Trailer 1	Trailer 2
Length (m)	14.6	8.2
Weight (kg)	6,260 (empty) 19,867 (loaded)*	2,948 (empty) 16,555 (loaded)
Trailer Moment of Inertia (kg/m^2)	150,194 (empty) 399,820 (loaded)	46,927 (empty) 130,522(loaded)
Dist. Between Trailer Tandem Axles (m)	1.52	1.22
Dist from CG to Ground (m)	1.65 (empty) 1.83 (loaded)	1.65 (empty) 1.83 (loaded)
Dist. from Trailer CG to Fifth Wheel (m)	7.66	3.94
Dist. from Trailer CG to Axle (m)	4.64	2.46
Half Lateral Dist. Between Centres of of Tire Contact on Trailer Axles (m)	1.03	1.03
Trailer Tandem Axle Load Transfer Coefficient	0.15	0.15

* - Payload = 13,607 kg (30,000 lb)

Table 3: Tire Data

Description	Value
Cornering Stiffness (N/deg)	4300
Peak Tire-Road Coefficient	0.47
Sliding Tire-Road Coefficient	0.25
Slip Corresponding to Peak Friction Coefficient	0.15

2.2 Simulation Manoeuvres

The constant radius turn and the lane change, were modeled and used as a platform for detecting the onset of jackknife and determining the performance of the anti-jackknife devices modeled.

The details of the two manoeuvres are as follows:

1) *Constant Radius Turn*

The radius of the turn was 86.7 m. The vehicles were simulated at speeds that equate to theoretical lateral accelerations of 0.1 g, 0.2 g and 0.3 g. The handling diagrams (discussed below) allowed for determination of the required input steering angle for each run. Braking to induce jackknife and trailer swing was applied through both wheel lock-up and varying brake distributions (Section 4.0) shortly after steady state was achieved.

2) *Lane Change Manoeuvre*

The lane change manoeuvre was simulated by inputting a maximum sinusoidal steer input of amplitude +5° to -5° over a time span of 2.2 seconds. Jackknife and trailer swing were induced through wheel lock-up at four different points in time. The first lock-up point was achieved just prior to initiating steer input. The second point occurred at the +5° peak of the steer input curve. The third point occurred when the steer input was zero (transition point from positive to negative). And the final lock-up point occurred at the -5° steer input peak.

2.3 Steady State Handling Characteristics

Steady state handling behaviour is an important aspect of vehicle handling studies. Handling diagrams in particular aid in determining any inherent instabilities of a particular combination, and give an idea as to the relative performance of the vehicles in each manoeuvre. Handling diagrams also enable the determination of the steer angle required to complete a desired steady-state manoeuvre, such as the constant radius turn. A more detailed discussion of steady state handling characteristics and handling diagrams is presented in our Stage 1 report (1).

The handling diagrams for the two configurations considered in this study are presented in Figures 1 and 2. The required steer input (δ) for a particular turn is determined by knowing the radius of the turn and the vehicle speed. The intersection of the appropriate constant velocity line and a vertical line on the "L/R" axis that corresponds to the turning radius produces a point (where L = tractor wheelbase and R = Turning Radius). The horizontal distance (constant lateral acceleration) between this point and the corresponding point on the "rL/U-δ" curve is the required steady state steer angle required to traverse the desired constant radius turn. Figure 1 illustrates the steer angle required for configuration 1 to traverse a 96.6 m radius turn at 0.2g.

3.0 JACKKNIFE DETECTION

The primary concern in jackknife prevention is sensing incipient jackknife at the earliest possible stage. The longer a jackknife situation is allowed to exist, the greater the instability becomes and hence the more difficult it becomes to control. Computer simulation results indicate that in a jackknife situation, 15 degrees of articulation, the generally accepted "point of no return", can be achieved in as little as 0.5 seconds from the time wheel lock-up occurs. At this point, the magnitude of the moment required to restrain articulation is very large (in the order of magnitude of 120,000 N•m for configuration 1). If a restraining moment of this magnitude is suddenly applied at this point in time, vehicle stability could be jeopardised. This became evident when a test simulation was conducted and it was observed that both the articulation acceleration and the tractor yaw acceleration doubled in magnitude and changed sign within only 1/10 of a second immediately following torque application.

Based on the discussion presented in the previous paragraph, it becomes evident that the detection of jackknife in its early stages is essential. Looking at this jackknife detection problem from a practical implementation point of view, it also becomes evident that a means of detection should be developed that utilizes measurable input parameters. One approach toward the sensing problem is depicted in Figure 3. In this flowchart, input parameters are monitored and used to determine theoretical outputs, which are then compared with measured output values. The input parameters to be measured may be driver controlled parameters such as velocity, steer input, and braking; and the measured output parameters may be articulation angle, articulation rate, tractor yaw angle, tractor yaw rate, lateral acceleration, or any other desired parameter. The next section describes some of the relations that were investigated in an attempt to relate values that may be recorded as inputs to measured values, in an attempt to determine if the vehicle is in a jackknife situation.

3.1 Simulation Results

15 deg/s of Tractor Yaw Rate: The first, and most simplistic, approach toward jackknife detection was to search for a single parameter which indicated that jackknife had occurred once a pre-determined value had been reached. This was also one of the approaches carried out in Stage 1, and was repeated here to examine the problem in more detail. In Stage 1, it was noted that 15 deg/s of tractor yaw rate appeared to be an early indicator of jackknife. Closer examination during this study revealed that this value could not always be used as an early indicator. Examination of Figure 4 shows that configuration 4 traversing a lane change manoeuvre at 57.5 km/h experiences tractor yaw rates in the order of a magnitude of $\pm$ 14 deg/s. One deg/s variance between this and the 15 deg/s value does not provide significant room to accommodate sensing error, or other configurations that may be more responsive than configuration 4.

Another example which disproved 15 deg/s of tractor yaw rate as an early jackknife indicator was the simulation run involving transversing a lane change manoeuvre at 33.2 km/h and locking the rear tractor tandem brakes just prior to initiating any steer input. In this case, when 15 deg/s of tractor yaw rate was reached, the articulation angle had already obtained a value of 17.5 degrees as shown by the shaded tractor in Figure 5. Thus, 15 deg/s would not even suffice as a late indicator as the configuration was well past the generally accepted value of 15 degrees of articulation that indicates a point of no control for jackknife.

Relationship Between Steer Angle and Articulation Angle: Looking at Figure 6, it would appear that a simple relationship exists between the steer angle at a given speed and the vehicle's angular characteristics for the constant radius turn. For example, both the articulation rate and the tractor yaw rate appear to experience transitions in direct relation to the steer input. However, examination of Figure 7 shows that a potential relationship may not be quite so straight forward for a stable manoeuvre such as the lane change. In this case, many of the articulation variables lead or lag the given steer input. This is most evident at time t = 2.7 seconds when the steer angle is zero and the articulation variables are something other than zero. At this point in time configuration 1 experiences a peak in articulation angle, while configuration 4 reached its peak prior to this time. An even more interesting condition occurs beyond this point, where the steer angle remains at zero and both the articulation angle and the articulation rate take a considerable amount of time (almost 2 seconds for configuration 1) to reach zero. Configuration 4, the more responsive configuration, displays a peak articulation rate of -8.74 deg/s, 1/10 of a second after the steer input has reached zero, and does not reach a zero value until over 1½ seconds have elapsed from the completion of steer input. Hence, due to these deviations, a simple relationship between the steer angle and the vehicle's articulation characteristics could not be found.

Lateral Acceleration: Another method of jackknife detection involved investigating variations in lateral acceleration. This approach was based on knowing that the lateral acceleration of the vehicle is equal to the vehicle velocity squared divided by the turning radius of the curve:

$$(a_n = \bar{V}^2/R) \qquad [1]$$

To physically implement this detection method, an accelerometer or gyroscope could be used to measure the existing lateral acceleration (a_n). This value could then be compared with the relation V^2/R, where V could easily be measured (speedometer, transmission or drive shaft rotational speed). This would leave the requirement of measuring a value for the turning radius, R. Obviously the turning radius is a function of the vehicle speed, the steer angle, and configuration parameters such as tractor and semitrailer wheelbase, tandem axles, dual tires, etc. [R= f(δ,V, Configuration Parameters)].

A simplistic approach toward determining the vehicle turning radius may be carried out by considering the geometric relationship between the tractor wheelbase and the steer angle, as shown in Figure 8. From this diagram it can be seen that the turning radius R can be approximated by:

$$R = L/\sin(\delta) \qquad [2]$$

substituting this value into equation [1] provides the following relationship:

$$a_n = \frac{\bar{V}^2 \cdot \sin(\delta)}{L} \qquad [3]$$

This simplistic relationship was tested against simulated values for lateral acceleration and a sample of the results are shown in Table 4. As can be seen by the values listed in the "% Diff." column, the relationship proved to be unreliable as values ranged from 2% to 980%.

Table 4: Comparison of Calculated Lateral Acceleration To Simulated Output

	TIME (s)	DELTA,δ (deg)	a_n Calc. (m/s^2)	a_n Output (m/s^2)	% Diff. (C-O)/O
Configuration 1 49.6 km/h Lane Change	0.72	2.80	1.48	1.44	2
	1.22	4.07	2.15	1.59	35
	1.62	-0.25	-0.13	-0.46	72
	1.92	-3.75	-1.98	-1.68	18
	2.62	-1.02	-0.54	-0.05	980
	3.00	0.00	0.00	-0.02	—
Configuration 4 57.5 km/h Lane Change	0.72	2.80	3.46	1.93	79
	1.22	4.07	5.03	3.20	57
	1.62	-0.25	-0.31	0.73	143
	1.92	-3.75	-4.63	-2.32	99
	2.62	-1.02	-1.26	-1.76	28
	3.00	0.00	0.00	-0.12	—

4.0 VARIATIONS IN BRAKE FORCE DISTRIBUTIONS

Our previous work had always induced tractor jackknife through lock-up of the rear tractor tires, and trailer swing through semitrailer tire lock-up. No other axle carried any braking power. The reason for this method of brake induction was to produce a worst case jackknife or trailer swing condition. Although this worst case condition was beneficial in studying the characteristics of jackknife, it presented a situation that would be relatively infrequent in practice. For these reasons, it was decided to study the characteristics of jackknife and the performance of the anti-jackknife devices as the brake force distribution per axle varied.

4.1 Brake Force Calculation

The braking distributions were expressed as B1:B2:B3, where B1, B2, B3 represented the ratios of the braking force of the front axle, tractor tandems and semitrailer tandems to the total vehicle braking force, all on a percentage basis.

To equate these ratios to applied forces, a few facts had to be known. Firstly, an upper limit on the maximum amount of braking force the vehicle could supply had to be determined; and secondly, the limit on the amount of braking force that could be supplied at each axle (or tandem) also had to be determined. Leucht, in his study of the directional dynamics of a tractor semitrailer during braking (2), stated that "In order to achieve some degree of compatibility between various tractors and trailers in the field, the trucking industry has adopted the philosophy that brakes should be sized on each axle to produce a braking force equal to 0.6 of its static rated loading at a standard air pressure of 60 psi". This philosophy was adopted for this study.

A sample calculation is best suited to explain how these ratios were used to determine the actual braking force at each axle.

Sample Calculation:

To determine the maximum braking force at each axle assuming a 20:40:40 braking distribution, and knowing:

$W = 100{,}000$ N = weight of vehicle
$F_1 = 18{,}000$ N = static load on tractor front axle
$F_2 = 40{,}000$ N = static load on tractor rear axle(s)
$F_3 = 42{,}000$ N = static load on semitrailer axle(s)

Maximum possible braking forces at each axle (tandem):

$$B1_{Max} = 0.6 \cdot F_1 = 10{,}800 \text{ N}$$
$$B2_{Max} = 0.6 \cdot F_2 = 24{,}000 \text{ N}$$
$$B3_{Max} = 0.6 \cdot F_3 = 25{,}200 \text{ N}$$

$B1_{Max}$ becomes limiting factor, therefore:

$$B1 = B1_{Max}$$
$$B2 = (40/20) \cdot B1_{Max} = B3$$

20:40:40 distribution becomes; 10,800:21,600:21,600 actual braking forces

4.2 Distributions Studied:

There were five different brake force distributions considered for this project, designated as D1 through D5, where:

D1	-	20:40:40
D2	-	33:33:33
D3	-	27:36:37
D4	-	15:45:40
D5	-	15:50:35

Common to each distribution is the rate at which a maximum brake force is attained. Research (3,4) showed that pneumatic systems, on the average, reach a line pressure of 60 psi in about 0.1 seconds. Thus, brake forces on each axle (or tandem) were ramped from 0 to a determined value in 0.4 seconds.

4.3 Simulation Results

D1 - 20:40:40: Figure 9 shows the results of this brake distribution being applied during a constant radius turn. Evident from the sequential positioning of the configuration is the loss of side force on all tires. The entire vehicle continued in the general direction of travel from the point where maximum brake force was applied. The slight increase in articulation (approximately 9°) indicated that the lateral forces on the tractor tandem tires were slightly more saturated than the other tires. This slight increase in the degree of saturation resulted in a mild jackknife situation.

D2 - 33:33:33: This ratio was investigated to determine the effects of having an equal braking force at each axle (or tandem). The results, shown in Figure 10, indicate that the reduction in braking force at the semitrailer tires (40% reduced to 33%) was significant enough to allow for the lateral forces to be maintained. The resulting jackknife action with this distribution was not as violent as the D1 distribution (20:40:40). The tire slip angles for the D2 distribution were approximately 12°, 10° and 2° for the tractor front, tractor rear and the semitrailer tires respectively, whereas they were 23°, 26° and 27° for the D1 distribution. Comparing Figure 9 with Figure 10 it can be seen that both configurations display approximately the same degree of articulation (within 2°). Brake distribution D1 (20:40:40) however, presents a worse situation since the entire vehicle, tractor and semitrailer, diverged from the intended path (evident from the large tire slip angles).

D3 - 27:36:37: This distribution approached ideal proportioning as was discussed in section 4.1 (i.e. 0.6 of static load) with only a slight variation. The static loading of the configuration is actually 17:39:44. For the D3 distribution a portion of the braking force was removed from the front axle and redistributed to the rear tractor axles and the semitrailer axles (3% and 7% respectively). This condition represented a situation that could realistically occur on the road.

Comparison of Figure 11 with Figure 9 will show that the position of the semitrailer in both diagrams is essentially identical, the only variance is in the yaw angle of the tractor. This yaw angle variance was due to the greater proportion of braking force applied on the rear tractor tires in distribution D1 over D3 which allowed the rear tractor tires to slide because of the greater reduction in lateral forces.

D4 - 15:45:40: The previous three distributions were useful exercises in investigating the importance of proper brake proportioning, but did not induce any significant jackknife instabilities. This distribution, D4, was simulated to investigate unstable jackknife situations that can occur with braking forces being applied at all axles. As can be seen from Figure 12, this distribution was derived from distribution D1 (20:40:40) where 5% of the total braking force was removed from the tractor front axle and redistributed to the tractor rear axles. This redistribution allowed for the front tires to maintain a larger portion of the available lateral restraining forces. Similarly, the increase in the magnitude of rear tractor tire braking forces (of D4 over D1) decreased the available amount of lateral tire forces on the rear tractor axles. The increase in lateral forces on the front tractor tires combined with the decrease in the lateral forces on the rear tractor tires increased the pivoting action of the tractor about the centre point on the front axle (jackknife action). Also evident from Figure 12 is a slight rotation of the semitrailer, which resulted from saturation of the lateral forces on the semitrailer tires due to braking.

D5 - 15:50:35: The D5 distribution was derived from D4. The difference between the two distributions is that 5% of the total available braking force was redistributed from the semitrailer tires to the rear tractor tires in an attempt to increase the jackknife instability. Figure 13 shows that the 5% increase in braking force at the rear tractor tires had almost no effect on the absolute yaw angle of the tractor. The reason why this increase had no effect on the jackknife action was because the D4 distribution (15:45:40) had already exceeded the maximum available braking force of the tractor tandems (45% of total), and any further increase would be insignificant. In fact, when 5% of the total available braking force was removed from the semitrailer tires, the semitrailer was able to track due to the increase in available lateral forces at the tire road interface.

Comparison of Figure 13 with Figure 12 showed that the articulation angle of the configuration in Figure 13 (D5) is larger than that in Figure 14. The variance in articulation is due entirely to the ability of the semitrailer to maintain significant lateral forces on the tires when subjected to the D5 distribution (15:50:30).

4.4 Summary

The results obtained from modelling the 5 brake force distrubutions distributions are best summed up by considering Figure 14. Figure 14(a) shows the articulation angle as time varies for each braking distribution and Figure 14(b) shows the tractor yaw angle as time varies. If jackknife can be classified solely by the degree of articulation, then braking distribution D5 would be considered as the worst jackknife situation. Recall from the previous section that the reason why braking distribution D5 articulated to a greater degree than D4 was because the reduction of braking force applied at the semitrailer tires was enough to allow for a significant increase in the lateral forces. Therefore, braking distribution D4 can be considered to cause a more "unstable" situation than distribution D5.

The remaining three distributions, D1, D2 and D3, each showed considerable improvement over the worst case jackknife situation. It is difficult to label which distribution is "better" than the other with the output available from the program (physical testing would be best suited for this purpose). Each plot, Figures 9, 10 and 11, shows the configuration essentially travelling in the direction of travel from the point of brake application. These plots can be compared with Figure A3 which illustrates the configuration traversing a constant radius turn without any form of brake application (ideal situation). Only distribution D3 (33:33:33) however, shows any indication of maintaining significant lateral forces at the semitrailer tires to allow the semitrailer to track.

5.0 VARIATIONS IN SURFACE FRICTION

While investigating the effects new braking distributions had on jackknife, it became evident that if the effects of varying braking conditions were to be modeled, varying friction conditions should also be examined. This endeavour (varying friction conditions) was beyond the scope of work, but was pursued because of its importance.

Presented in Figure 15 are plots of brake force coefficient (friction coefficent) as wheel slip varies (μ/slip curves) which was originally presented in a report prepared by T E S Limited for Transport Canada in March 1989 (5). These curves (which are representative of curves presented in References 6 and 7) are functions of many variables, and as a result, are difficult to produce definitively, however, the available literature shows general agreement on certain points as follows:

- The peak value μ_p which can be achieved is typically reached in the region of 10% to 20% wheel slip.
- The locked wheel (100% slip) value μ_s which is achieved when braking torque exceeds the available friction force, may vary from μ_p.
- For high coefficient surfaces, such as dry concrete or dry asphalt, there is a decrease of 10% to 15% from μ_p to μ_s. That is, μ_s/μ_p is typically 0.85 to 0.90. (Note, however, that some papers show this decrease as effectively 0% with $\mu_s/\mu_p = 1.00$).
- For wet pavement, a large decrease occurs from μ_p to μ_s with factors of 0.60 to 0.80 being common.
- For low coefficient surfaces, such as snow and ice, there is effectively no difference between μ_p and μ_s.
- For "fresh" snow, and gravel surfaces, there is actually an increase from "μ_p" to μ_s of 20% to 30%.

All computer simulations carried out in Stage 1, and the bulk of simulations conducted in Stage 2, utilized friction coefficients of 0.47 and 0.25 for the peak and sliding values respectively. As can be seen from Figure 15, these values do not represent a commonly occurring situation. Since the forces at the tire/road interface are one of the major factors that dictate the performance of a vehicle, an investigation into the effects of varying friction values became important.

5.1 Simulation Results

The computer simulation model makes use of a user defined μ/slip curve through a simplistic 3 point representation. Four different μ/slip curves that were used in various simulation runs are presented in Figure 16.

Similar to the braking distribution naming convention, the four curves were labelled as:

U1	-	Snow (μ_p = 0.27, μ_s = 0.26)
U2	-	Wet Pavement (μ_p = 0.60, μ_s = 0.35)
U3	-	Trial 1 (μ_p = 0.47, μ_s = 0.46)
U4	-	Trial 2 (μ_p = 0.77, μ_s = 0.76)
Original	-	(μ_p = 0.47, μ_s = 0.25)

The following sections describe the results obtained when these μ/slip curves were simulated.

U1 - Snow (μ_p = 0.27, μ_s = 0.26): Since our investigation mainly dealt with jackknife induced through wheel lock-up, μ_s became the predominant friction value. Further investigation into low friction values without the occurrence of a larger peak value (μ_p) became a logical next step, and snow was therefore simulated. Figure 17 pictorially represents the results of a brake induced jackknife (wheels locked at the rear tractor tandems) of configuration 1 with the road surface being simulated as snow covered. As can be seen, a significant jackknife situation exists, but comparison with Figure A2 shows that a similar simulation with the original μ/slip curve (μ_p = 0.47, μ_s = 0.25) produced a much more severe jackknife situation.

U2 - Wet Pavement (μ_p = 0.60, μ_s = 0.35): To cover the full spectrum of maximum and minimum friction values, a realistic μ/slip curve on the upper end of the spectrum was next chosen for analysis. Wet pavement was chosen to be simulated because of the spread between μ_p and μ_s. Figure 18 shows the results of the simulation.

Comparison of Figure 18 with Figure 17, at similar points in time, reveals that U2 produced greater jackknife instabilities than U1. When Figure 18 is compared to Figure A2, the original μ/slip curve (μ_p = 0.47, μ_s = 0.25), no differences can be detected when compared at the same point in time. Surprisingly, the larger friction coefficients which theoretically are capable of producing larger forces, result in a jackknife situation worse than U1. This result led to the hypothesis that the friction factor that most significantly affects jackknife is not the magnitude of the friction coefficients, but the size of the spread between μ_p and μ_s. U3 and U4 were simulated to reinforce this theory.

U3 - Trial 1 (μ_p = 0.47, μ_s = 0.46): As was just previously mentioned, this μ/slip curve was simulated to investigate the effect altering the gap between μ_p and μ_s, at varying relative magnitudes, had on the resulting jackknife. Figure 19 shows the results of this simulation. The jackknife situation which resulted from this simulation was milder than the original μ/slip curve (μ_p = 0.47, μ_s = 0.25, Refer to Figure A2). In fact, the jackknife action was even milder than U1 (μ_p = 0.27, μ_s = 0.26) (refer to Figure 17). This result was not surprising since the difference between

μ_p and μ_s for both curves were the same, only the relative magnitudes varied. Since the values of μ_p and μ_s for U3 were relatively large when compared to U1, the tires were able to supply larger side forces which reduced the jackknife action.

U4 - Trial 2 (μ_p = 0.77, μ_s = 0.76): This final μ/slip curve was modeled to verify that reducing the gap between μ_p and μ_s had a direct impact on reducing the severity of jackknife action. Although the nominal 0.01 difference between μ_p and μ_s does not realistically occur at these friction levels, it was modeled solely for the purpose of exploring the effects the gap between μ_p and μ_s had on jackknife. Obvious from Figure 20 is the fact that U4 produced the mildest jackknife when compared to all other μ/slip curve simulations (U1, U2, U3 and the Original), which reinforces this point.

5.2 Summary

The articulation angle and tractor yaw angle versus time for the four μ/slip curves are presented in Figure 21. U2 (μ_p = 0.60, μ_s = 0.35) and the original friction values (μ_p = 0.47, μ_s = 0.25) displayed identical characteristics up until time t = 3.06 seconds where the vehicle approached the roll-over condition (a stopping criterion for the program) for the U2 friction values. Obvious from the figure is the fact that these two μ/slip curves result in the vehicle experiencing the worst jackknife. Recall that U2 and the original μ/slip curve were the only two curves with a significant gap between μ_p and μ_s ($\approx$0.23).

The remaining three distributions all possesed a 0.01 variance between μ_p and μ_s. The jackknife action improved as the relative magnitude of the friction values became larger. Unfortunately, as the friction values increased, so did the tendency of the vehicle to approach the roll-over condition as may be seen from the termination point of each curve in Figure 21. It should be noted that the program may have been over responsive when determining roll-over conditions.

6.0 ACTIVE ANTI-JACKKNIFE DEVICE

By definition, an active anti-jackknife device should "cause or promote action or change". This 'action' would typically take place immediately following the detection of the occurrence of jackknife. Since our earlier investigation into the detection of jackknife (Section 3.0) did not provide a solution to the problem, the anti-jackknife device was still modeled with the assumption that a detection mechanism was in place. For the purpose of simulation, this detection mechanism simply consisted of indicating that a jackknife situation had just occurred when the lateral tire forces at the rear tractor tires suddenly decreased by a value of 1000 N.

The restraining moment was applied in such a way so as to redistribute, to the trailer, the increment of tractor angular momentum associated with the incipient jackknife. Specifically, a calculated torque was applied about the fifth wheel that reduced the tractor yaw acceleration to zero. Figure 22 illustrates the concept of how this torque was calculated. The restraining moment must counter balance any moment caused by the tractor yaw acceleration ($I \cdot \Psi$) and any additional moment that may be caused by the force vector "ma". Thus, as illustrated in the diagram:

$$T = I \cdot \Psi + m \cdot a \cdot D \qquad [4]$$

Where
- T = Restraining Torque
- I = Tractor Moment of Inertia
- Ψ = Tractor Yaw Acceleration
- m = Mass of Tractor
- a = Tractor Acceleration
- D = Moment Arm of 'ma' Vector

6.1 Simulation Results

Figure 23 presents the simulation results of configuration 4 involved in a brake induced jackknife situation on a constant radius turn with the active anti-jackknife device acting to restrain any unwanted increase in articulation. Comparison of this diagram with Figures A3 and A4 shows that the device performed extremely well. Two seconds after wheel lock-up occurred, the articulation angle was only -17.5 degrees, as opposed to -84.5 degrees after 1.5 seconds for the unrestrained situation.

A more involved examination of the device's performance can be obtained from the information presented in Figure 24. Figure 24 (b) shows the magnitude and time history of the restraining torque applied about the fifth wheel of the configuration. The sudden peaks in restraining torque illustrate a somewhat undesirable condition. The effects of these sudden peaks can be seen in the articulation and tractor yaw characteristics presented in 24 (a). These sudden changes in articulation and tractor yaw tend to indicate that stability problems may be inherent with this device. Since the program was not designed to analyze stability problems in detail, this observation is only speculation.

As was previously mentioned, physical testing is best suited for stability analysis. This point is reinforced by the results obtained by the Ministry of Transportation and Communication (Ontario) during their two part project aimed at evaluating several anti-jackknife devices (8) and validating the computer simulation model (9) which was used as the main analytical tool in this project. The applicable portion of the project involved the Safe-T-King anti-jackknife device. During their testing phase, one particular manoeuvre forced the vehicle to experience stability problems due to a 'bouncing' action caused by the Safe-T-King device. When the same manoeuvre was simulated on computer, the over responsiveness of the model predicted a somewhat different result.

Aside from these potential stability problems, the device performed extremely well. Simulations involving Configuration 1 under the same circumstances showed very similar results. Articulation for Configuration 1, however, reached -23.5 degrees because an upper limit of $\pm$ 100,000 N•m was programmed into the active device module. This limit was included because it represented a potential maximum value attainable.

7.0 PASSIVE ANTI-JACKKNIFE DEVICES

A passive device, by definition, is a device which is "acted upon, affected, or influenced by something external". A passive anti-jackknife device would therefore apply a restraining moment based on a set of existing conditions. There were five different types of passive anti-jackknife devices examined in this study. Each device had a constant, or set of constants, that could be varied to subtly change the resulting restraining moment. Figure 25 quickly outlines; which devices were studied, their characteristic equation, the simple mechanical component the devices were based on, and the set of constants that were used to fine tune their performance.

7.1 Spring Type Device

A method of controlling jackknife that increased the applied restraining torque as articulation increased was desired. The spring, with characteristic equation:

$$F = -K \cdot x \qquad [5]$$

does exactly this and was therefore used as a base component for the device. The torque equation listed in Figure 25:

$$T = K \cdot \mathrm{Tan}(\Gamma) \cdot D^2 \cdot (\text{Sign of } \Gamma) \qquad [6]$$

where, Γ = articulation angle, is simply a geometric adaptation of equation [5]. The equation was formulated such that as the configuration articulated the restraining moment would increase.

Figures 26, 27 and 28 show configuration 1 in a constant radius turn subjected to jackknife (through wheel lock-up) with the spring type anti-jackknife device enabled in each case. The spring constant in Figure 26 is 50,000 N/m, 75,00 N/m in Figure 27 and 100,000 N/m in Figure 28. Comparison of Figure 26 with Figures A1 and A2 of Appendix A shows that the jackknife situation was slightly improved with a spring constant of 50,000 N/m. Increasing the constant even more improved the situation as can be seen by the position of the configuration in Figure 27.

When an even greater increase in spring constant was simulated the configuration (as shown in Figure 28) for some reason resulted in a worse situation. It could be speculated that the torque induced by the spring restraining vehicle articulation was sufficient enough to rotate the semitrailer, but due to the mass and size of the semitrailer this would be unlikely. Simple static calculations indicated that the amount of torque necessary to rotate the tractor about the fifth wheel is over five times that required to rotate the semitrailer about the fith wheel. Although momentum was not considered in the above calculation, it would seem that a semitrailer of this given weight and size should be able to handle any excessive forces the unstable tractor may induce. Another point that raises some questions concerning this particular simulation run is the

value of the tire slip angles that were produced as output. The magnitude of the slip angles on all tractor tires were in the order of 20,000 degrees and 125 degrees on the semitrailer tires. A detailed investigation into how these slip angles were handled within the program was not carried out as it was deemed to be well beyond the scope of work.

In summary, the spring type anti-jackknife device did show improvement over the unrestrained jackknife situation shown in Figure A2 (excluding the simulation where K = 100,000 N/m,). Figures 29 and 30 show the three variations on this type of device. An improvement in the vehicle performance is obvious as the spring constat was increased from 50,000 N/m, 1A, to 75,000 N/m, 1B, but a severe regression occurred when a further increase to 100,000 N/m, 1C, was simulated.

7.2 Linear Damping Device

The linear damping, or dashpot, device was modeled to investigate the effects mild damping would have on jackknife. As can be seen from Figure 25, the basic mechanical component of the device is the dashpot, which has a characteristic equation of:

$$F = -C \cdot V \qquad [7]$$

similar to the spring type device, this basic equation was used as the fundamental building block for the anti-jackknife device. The linear velocity component, V, was expressed in terms of articulation by substituting:

$$V = \dot{\Gamma} \cdot D \qquad [8]$$

into equation [7] (where D = distance from the pivot point to the point where the damper acts). This force component was then multiplied by the moment arm D to obtain the equation for the restraining torque:

$$T = C \cdot \dot{\Gamma} \cdot D^2 \qquad [9]$$

Three coefficients of viscous damping, C, were simulated, namely 100,000, 150,000 and 200,000 N•s/m. None of these simulated devices showed considerable improvement, and were therefore not investigated any further. The results of the C = 200,000 N•s/m simulation is presented in Figure 31. Notice how the trailer swings in a manner similar to the spring type device in Section 7.1. This situation arose during all simulations with this device.

Figure 32 displays the jackknife characteristics of configuration 1 outfitted with the linear damping device. Obvious from the graphs is the fact that both articulation and tractor yaw angle were better controlled as the viscous damping coefficient was increased. Figure 33 shows how the applied restraining moment increased as the damping coefficient increased.

7.3 Spring/Dashpot System

Both the spring type device and the linear damping device showed areas that required improvement. The spring type device, as can be seen in Figure 29, showed a mild form of low frequency "bouncing". The linear damping device, evident from Figure 33, was unable to supply restraining torque at an adequate rate to effectively control the occurring jackknife. The two devices, spring type and linear damping, were combined to form a spring/dashpot system in an attempt to improve the stand alone performance of the individual devices. Figure 25 presents the equation used to model this device, and values of the constants used to vary the rate and magnitude of the applied torque.

The overall performance of this device did not show significant improvement over its two stand alone component devices (spring type or linear damping). In fact, regardless of the combination of constants used, the performance of the spring device 1B (K = 75,000 N/m) could not be matched. Figure 34 shows a sample plot of device 3E (K = 50,000 N/m, C = 100,000 N•s/m). Notice how the semitrailer, once again, rotates with the tractor.

A summary of the results obtained with this device are shown in Figures 35 and 36. Spring/Dashpot device 3B (K = 100,000 N/m, C = 100,000 N•s/m) improved the vehicle performance to the greatest degree (compared to the others), and 3C (k = 50,000 N/m, C = 50,000 N•s/m) showed the least amount of improvment.

7.4 Contant Damping Device

The constant damping device, as can be seen from Figure 25, was based on the friction that occurs between two surfaces in contact. The characteristic equation:

$$F = \mu \cdot N \qquad [10]$$

was used as a basis for developing the device. As illustrated in Figure 25, a brake caliper system can be used to physically achieve the desired results of this device, where N, the normal force, can easily be varied. Five different normal forces were simulated, and the friction coefficient was kept constant at 0.5 which is a typical value of brake pads on steel.

Figures 37 and 38 show the results of configuration 4 attemtping to traverse a constant radius turn when jackknife was induced with the constant damping device enabled. In Figure 37 the normal force was 80,000 N, 4C, and in Figure 38 the normal force was 120,000 N, 4D. Both improved the vehicle performance, as can be seen through comparison of Figure A4. Constant damping device 4D reduced the jackknife action (basically nullified it) considerable more than devices 4A, 4B and 4C, however, the vehicle was forced to deviate from its intended path (Figure A3) by over seven metres. Although jackknife was essentially elminated, vehicle controllability may have been sacrificed. Figure 39(a) shows how

device 4E has effectively reduced articulation to zero. The tractor yaw angle with device 4E, as shown in Figure 39(b), was reduced to a level less than the ideal situation. These results lead to the speculation that an applied constant torque of this magnitude, for this size of configuration, would produce a situation that is similar to driving a long multi-axled straight truck (i.e. articulation is continuously restricted to a minimum). To further examine the loss of control caused by this device, configuration 4 was simulated to manoeuvre a lane change with the device engaged. The results are shown in Figure 40. Comparison of Figure 40 with Figure A5 shows a y-coordinate deviation of approximately two metres, which clearly indicates the device's inability to allow for proper vehicle tracking.

A final concern involved the device's ability to improve vehicle performance in jackknife situations with larger tractor semitrailer combinations. Figure 41 shows the results of device 4C (N = 80,000 N) used to control jackknife of configuration 1 on a constant radius turn. There is a definite improvement over the worst case situation, Figure A2, but not as significant as when applied to configuration 4, Figure 37. The obvious conclusion to be drawn is that with jackknife situations involving larger configurations (and thus larger impending forces), larger restraining torques must be applied to achieve results similar to smaller configurations.

7.5 Variable Damping Device

The final passive type device studied was the Variable Damping Device. Figure 25 shows the equation that describes the device, and the list of constants that were used to investigate the performance characteristics of the device.

Variable damping was based on fluid damping, which is achieved by restricting or controlling the flow of a fluid. There are two forms of fluid damping; viscous damping, which occurs under laminar flow conditions and is proportional to velocity, and turbulent flow damping which is proportional to velocity squared. Our variable damping device, as can be seen from the equation presented in Figure 25, was derived to form a modified version of velocity squared damping. In this case, an additional multiplying factor "K2•Γ" was included to increase the applied restraining torque as articulation progressed.

Figures 42 and 43 show the results of configuration 4 in a jackknife situation on a constant radius turn enabled with variable damping devices 5A and 5G respectively. Both devices showed considerable improvement over the unrestrained situation presented in Figure A4. Device 5G (K1 = 900,000, K2 = 10) allowed the configuration to articulate to a maximum of 25 degrees, but only after providing an additional 0.8 seconds before achieving 15 degrees of articulation, over the unrestrained situation. Within 1.5 seconds from point of wheel lock, device 5G kept the articulation angle to less than 16 degrees, which is an immense improvement over 70 degrees achieved in the unrestrained case.

The amount of deviation from the intended path is approximately 1 m for device 5G (compare Figures 43 and A4). This device and the active device, which followed an almost identical path, had minimal deviations from the ideal path. The variable damping device, however, displayed much more stable characteristics as can be seen from Figure 44. Recall Figure 24 where the articulation characteristics experienced sudden changes when the active anti-jackknife device was enabled.

Another interesting feature of this device was the method in which restraining torque was applied during jackknife. As can be seen from Figure 45, the maximum value of restraining torque applied essentially does not vary, regardless of the value of the damping constant (surprisingly the largest damping constant produces the lowest torque). The only significant variance occurs with the rate at which the torque us applied. Device 5G (K1 = 900,000) reaches its maximum value approximately 0.2 seconds prior to device 5A (K1 = 100,000). This negligible variance in the magnitude of applied restraining torque and small variance in response rate produces a significant improvement in the performance of the vehicle (recall Figures 42 and 43).

Unlike the constant damping device where an increase in the constants of the equation greatly reduced the manoeuvrability of the vehicle, a similar increase in the constants of the variable damping device appeared to have a negligible effect. This condition is illustrated in Figure 46 which shows considerable improvement over the constant torque device (Figure 40) when a lane change manoeuvre was attempted, and did not vary from the ideal case (Figure A5).

The device's performance on configuration 1 was essentially the same, with the single exception that the magnitude of the restraining torque reached a maximum value of approximately 83,000 N•m which was approximately twice as large as the torque applied in configuration 4. Figure 47 shows configuration 1 in a jackknife situation on a constant radius turn with variable damping device 5G enabled.

When the variable damping device was simulated in a jackknife situation with one of the braking distributions previously discussed, or under different surface friction values (varying μ/slip curves) an interesting observation was made. Regardless of the method jackknife was induced, and the rate at which it occurred, the variable damping device performed in a consistent manner. This is easily explained by considering the fundamental parameter upon which the variable damping device is based; articulation rate. As was discussed in Sections 4 and 5, variations in either brake application or surface friction generally produced a less severe jackknife than was normally simulated (μ_p = 0.47, μ_s = 0.25, locked tractor tandems). Since the jackknife was not as severe (lower rates) the restraining moment of the variable damping device was comparatively lower (due to its direct dependency on articulation rate). Thus, it would appear that regardless of the conditions that cause jackknife, the performance of the variable damping device would remain consistent.

7.6 Summary

Five passive type anti-jackknife devices were studied; the spring type device, the linear damping device, the spring/dashpot system, the constant damping device, and the variable damping device. The first three, spring type, linear damping and spring/dashpot system, generally showed improvement over the unrestrained jackknife situation, but not to a satisfactory degree. In a few instances, the devices appeared to induce an unstable swinging motion of the semitrailer. It is suspected that this peculiar motion was due to problems associated with the method in which the "simulation program" handled the tire slip angles (as was discussed in Section 7.1).

The remaining two passive type devices; the constant damping and variable damping, showed considerable improvement over the others. In fact, the constant damping device totally eliminated all jackknife action in one instance. Unfortunately, vehicle controllability was significantly hampered under normal driving manoeuvres such as the lane change because the restraining moment was continuously applied from time t = 0. The variable damping device appeared to out perform all the other devices. It managed to control the path of the configuration as effectively as the active type anti-jackknife device, but in a more stable manner. Although articulation was not restrained as well as it was with the constant damping device, the variable damping device allowed the vehicle to track closer to the intended path. It also showed further merits over the constant damping device by providing much greater vehicle control to the driver during normal manoeuvres. In the 1.5 second time window from the point of wheel lock (a conservative night time driver reaction time) the variable damping device held vehicle articulation to less than 16 degrees, which we speculate, would allow a driver a good chance to regain control of his/her vehicle.

8.0 ANTI-LOCK BRAKE SYSTEM (ABS) CONSIDERATIONS

The study of anti-lock brake systems (ABS) was well beyond the scope of this project, and is included only to provide some insight as to how an anti-jackknife system (AJS) will interact with ABS. As ABS was not researched and studied, only a simplistic presentation of its theory is provided.

Anti-lock brake systems attempt to maintain tire side forces while braking and hence eliminate the major cause of jackknife. The amount of wheel slip is controlled, thus preventing wheel lock-up. Wheel slip is the ratio of the actual rotational wheel speed to the free rolling wheel speed. Thus, zero slip is the free rolling condition and lock-up is 100% slip. As the wheel moves from the free rolling to the locked condition, the side forces generated at the tire/road interface decrease rapidly as shown in Figure 48. Similarly, the braking force is at a maximum in the 10% to 30% wheel slip region. Below 10% slip the braking forces drop off rapidly and are not readily controlled. Above 30% slip the tire side forces are too small for good lateral stability. Anti-lock brake systems, therefore, are designed to operate approximately within this region to provide as much lateral stability as possible without adversely affecting the straight line braking performance.

It is anticipated that there are three situations in which jackknife can occur, even with ABS on board and functioning. They are; very aggressive steer inputs, power jackknife, and braking when the lateral forces at the rear tractor tires are almost saturated.

8.1 Jackknife Without Braking

Aggressive Steer Inputs - Mikulcik, in his SAE paper (10), addresses the problem of jackknife induced through aggressive steer inputs. He states that jackknifing of the vehicle occurs when the slip angles of the rear tractor wheels reach a value corresponding to the maximum side force. This can be explained by realizing that during a steady-state turn there exists, at each tire, side forces which act towards the inside of the turn and have relative magnitudes in the proportions required for a constant yaw rate. Any change in this balance of side forces allows the rate to change. For the configuration used in his study, Mikulcik found that step steering input angles of 0.06 radians (3.44 degrees) and greater produce a non-linear jackknife action.

For the aggressive step-steer inputs, 0.06 radians and above, as stated by Mikulcik, the rear tractor tire side forces build up until they reach the limiting values corresponding to slip angles which result in maximum possible side force. Limiting of the side forces at the rear wheels corresponds to limiting of the yaw moment available for keeping the tractor headed in the direction of travel. Any increase beyond this limit increases the yaw rate of the tractor which indicates the beginning of jackknife. Thus, it is possible to achieve jackknife in this situation without inducing any form of braking. In an instance such as this,

the combined systems of ABS and AJS can only compliment each other, where the AJS would be able to restrain jackknife action when the ABS could not.

Power Jackknife - Tractor jackknife can be caused by too much engine torque transmitted to the drive axles, such as when gearing down, or applying excessive acceleration on an icy surface. This form of jackknife is known as a power jackknife. Although it can be just as dangerous, it is generally not as frequent except on mountainous or hilly terrain.

Since the computer simulation program used in this project is based on a specified initial constant velocity, it is impossible to simulate a power jackknife situation. However, the power jackknife situation exists because of the saturation of lateral forces at rear tractor tires. This loss of the lateral forces can be parallelled to the loss associated with rear tractor tire lock-up induced by braking.

Power jackknife situations typically occur at low speeds. Our work showed that the severity of jackknife increases as the speed of the vehicle increases. A power jackknife would therefore show similar, but less sever, jackknife characteristics than the brake induced situations previously discussed.

8.2 Brake Induced Jackknife

Braking with ABS - In theory, and under ideal circumstances, ABS should be able to completely eliminate any brake induced forms of jackknife. It is speculated, however, that there is a situation in which jackknife may occur with a tractor semitrailer outfitted with ABS. The situation involves braking when the forces induced by the lateral acceleration of the tractor semitrailer approach the saturation limit of the lateral restraining forces at the rear tractor tires (i.e. cornering on a sharp turn at a relatively high velocity).

As was discussed in Section 8.0, anti-lock brake systems operate typically in the 10% to 30% slip range. When a vehicle is operating near its lateral restraining force limit, any imbalance due to braking may be significant enough to saturate these lateral restraining forces. Thus, under extreme circumstances, a brake induced jackknife may occur with a tractor semitrailer outfitted with ABS. The severity of the jackknife will depend directly on the effectiveness of the anti-lock brake system (i.e. on its ability to keep wheel slip within the 10% to 30% range). Figure 48 illustrates just how important it is for an anti-lock brake system to keep wheel slip within this range (ideally, towards the low end of the range). At 10% slip the tire side force is reduced to 76% of its free rolling value. At 30% it is reduced to only 46% of its free rolling value. It is therefore theoretically possible to have a brake induced jackknife occur under these conditions. The jackknife will, however, be very mild in comparison to rear tractor tire lock-up.

9.0 CONCLUSIONS

1. From a practical sensing point of view, a method that can be used to detect the onset of jackknife for all configurations under any set of conditions was not found.

2. Investigation into various braking distributions showed that:

   - As the braking forces on the rear tractor axle(s) increase, the jackknife instability also increases.
   - Increasing the Tractor Front axle braking force reduces the jackknife instability.
   - Brake induced jackknife can be reduced with proper brake proportioning.

3. Jackknife instability increases as the difference between the peak and sliding friction coefficients increase.

4. An active type anti-jackknife device was successfully modelled and showed considerable merit in controlling jackknife. However, sudden changes in articulation characteristics indicate that vehicle stability may not be totally maintained. Physical testing is the most suitable means in which to answer these vehicle stability questions.

5. Five passive type anti-jackknife devices were simulated, and all showed merit in improving jackknife stability. The variable damping device showed superiority over the other passive devices by: controlling articulation in a seemingly stable manner; behaving in a consistent manner regardless of the impending conditions; and, having minimal effect on normal transient manoeuvres.

6. Three situations were identified where jackknife may occur with ABS functioning on a tractor semitrailer. They were; very aggressive steer inputs, power jackknife and braking when the lateral forces of the rear tractor tires are almost saturated. The first two cases, power jackknife and aggressive steer inputs, present situations which ABS is not designed to control. However, ABS would be of assistance in helping the driver regain control of the vehicle by reducing the possibility of wheel lock-up from subsequent braking actions. In this case, an anti-jackknife system would assist in controlling an adverse situation which the anti-lock brake system was unable to control. The final case, braking when the rear tractor tires are near saturation, presents a situation where AJS may complement ABS by controlling vehicle actions at the critical point where the lateral restraining forces are very sensitive to ABS performance.

7. The effectiveness of an anti-jackknife device is best measured against its ability to assist the driver with control of the vehicle. In the simulation model used, a pre-programmed steer input is continually being carried out regardless of the existing situation, and any assistance a driver may provide cannot be taken into account. The full effectiveness of an anti-jackknife device is therefore difficult to measure with this simulation model.

8. It was observed that in extreme jackknife circumstances the accuracy of the simulation program may be questionable. In particular, it was observed that the tire slip angles in these extreme cases reached values in the order of magnitude of thousands of degrees. How these large slip angles are handled within the code of the program was not investigated in great detail, as this was beyond the scope of the project.

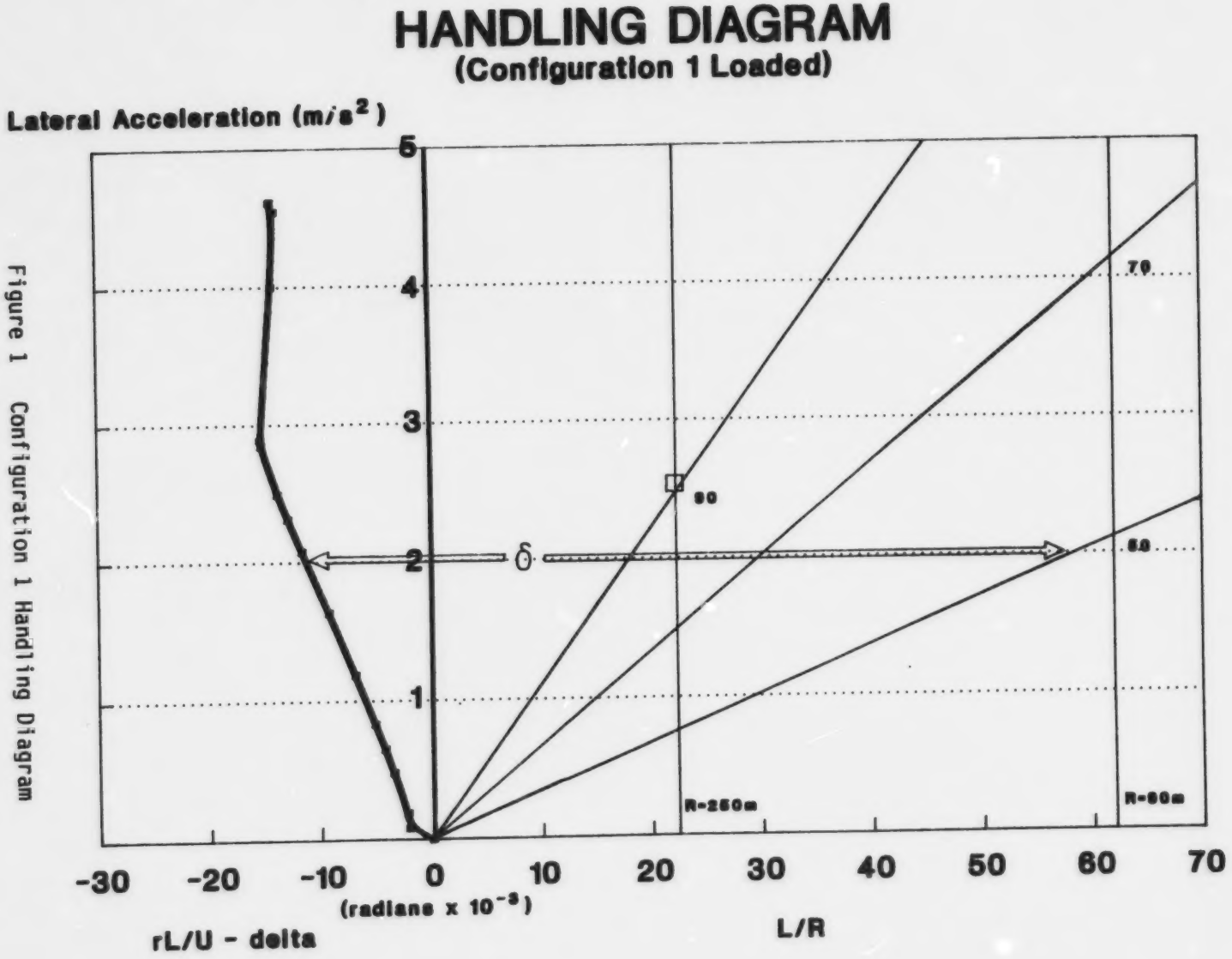


Figure 1 Configuration 1 Handling Diagram

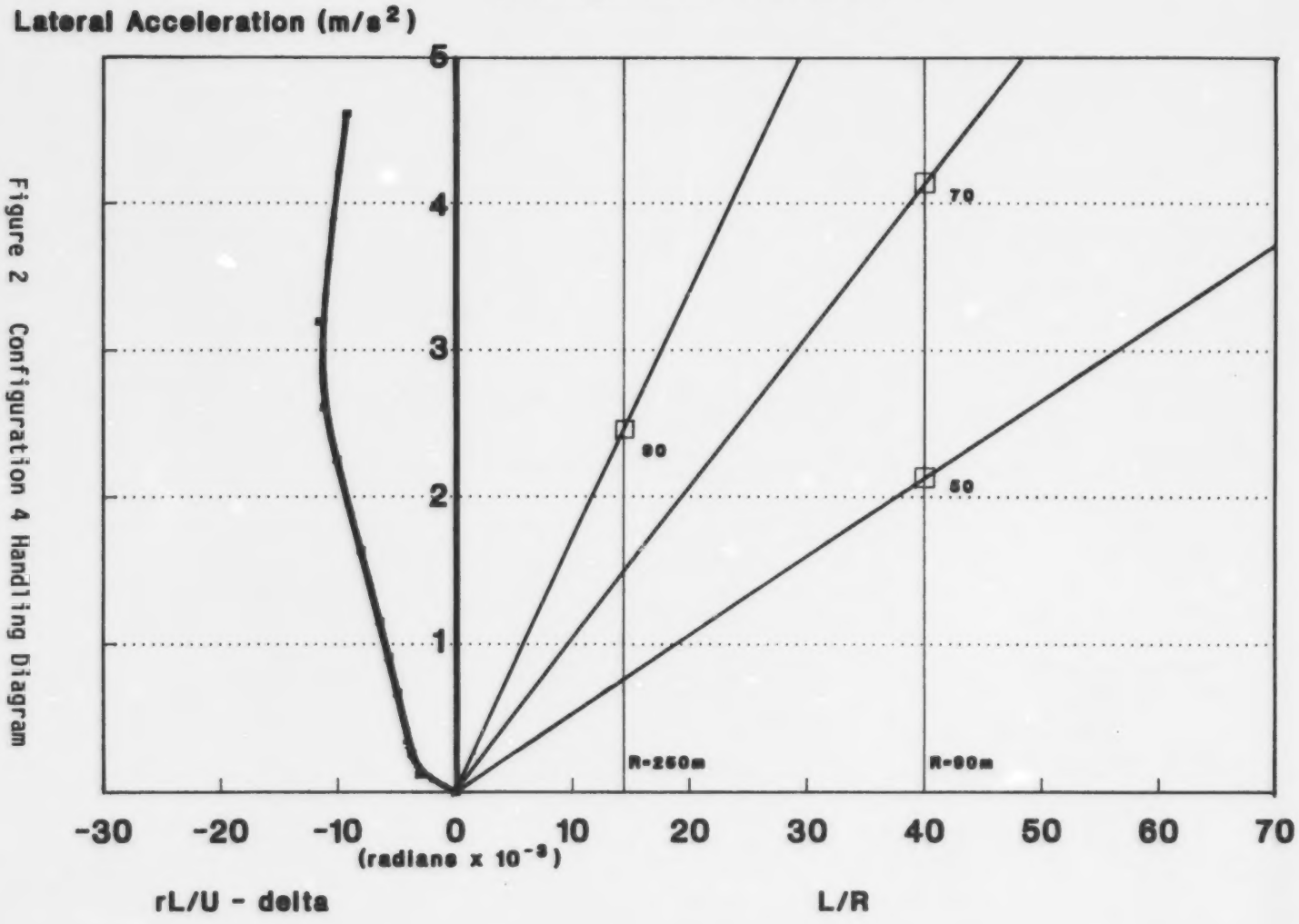


Figure 2 Configuration 4 Handling Diagram

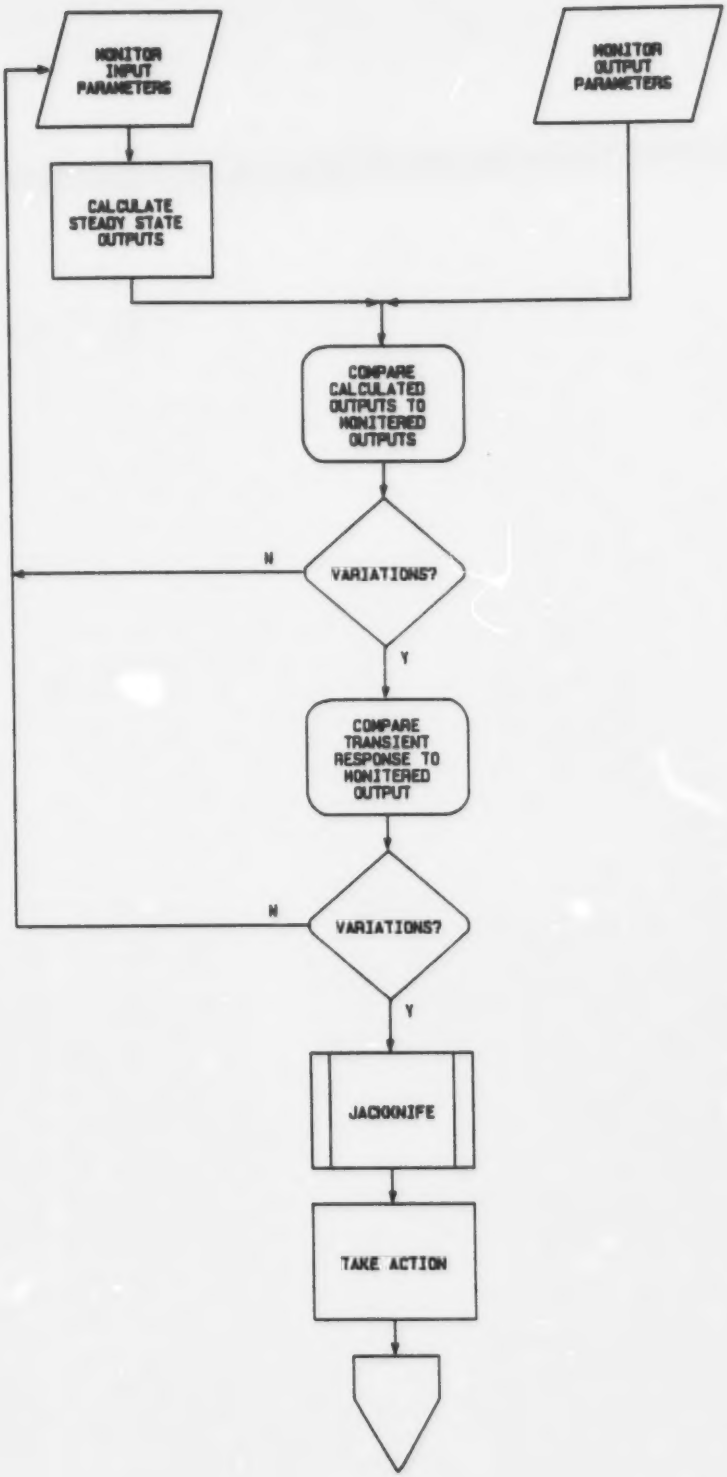


Figure 3 Jackknife Detection Methodology

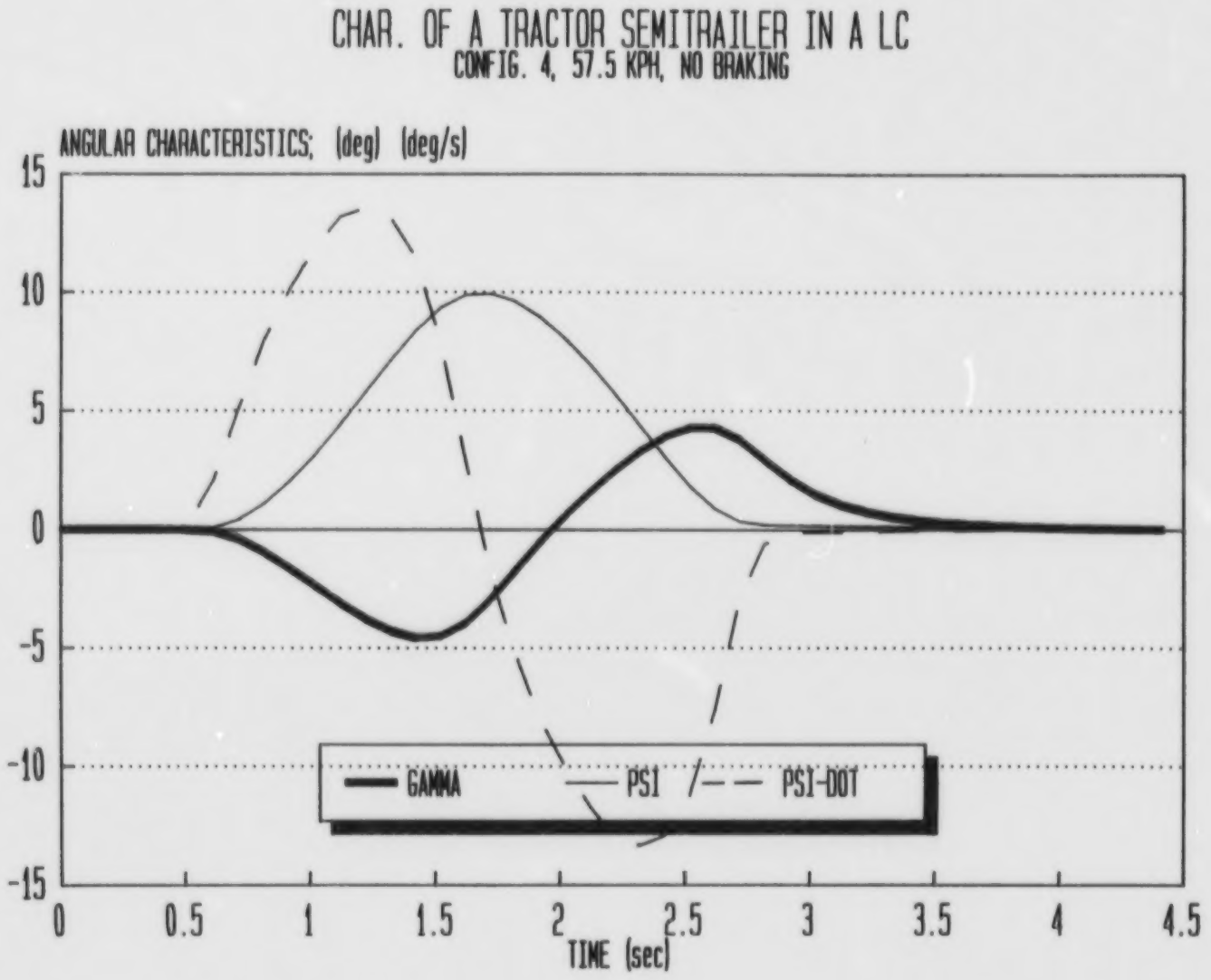
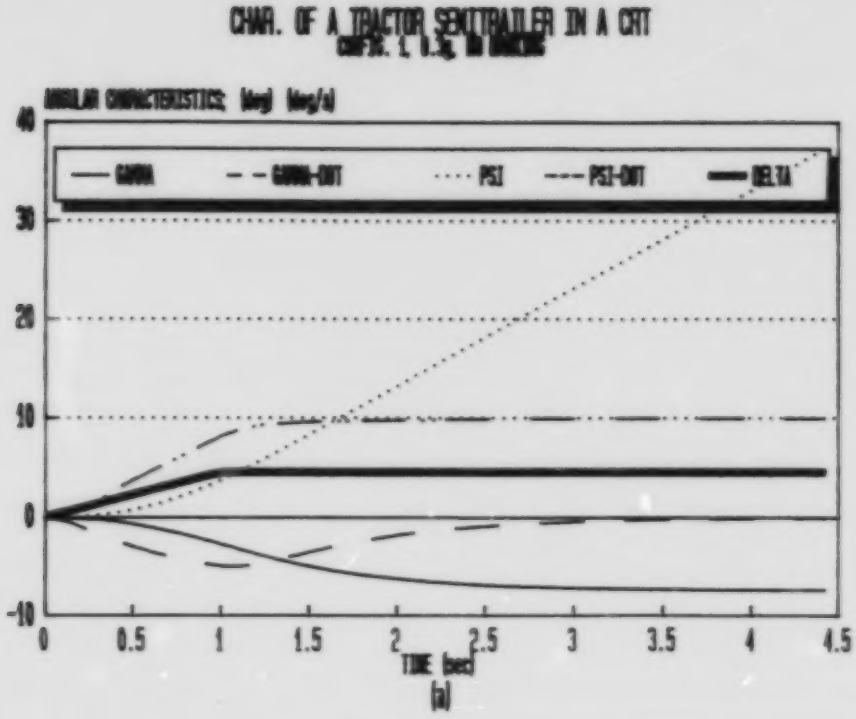


Figure 4 Characteristics of Configuration 4 in a Lane Change Manoeuvre

Figure 5 Configuration 1, Braking Before a Lane Change Manoeuvre



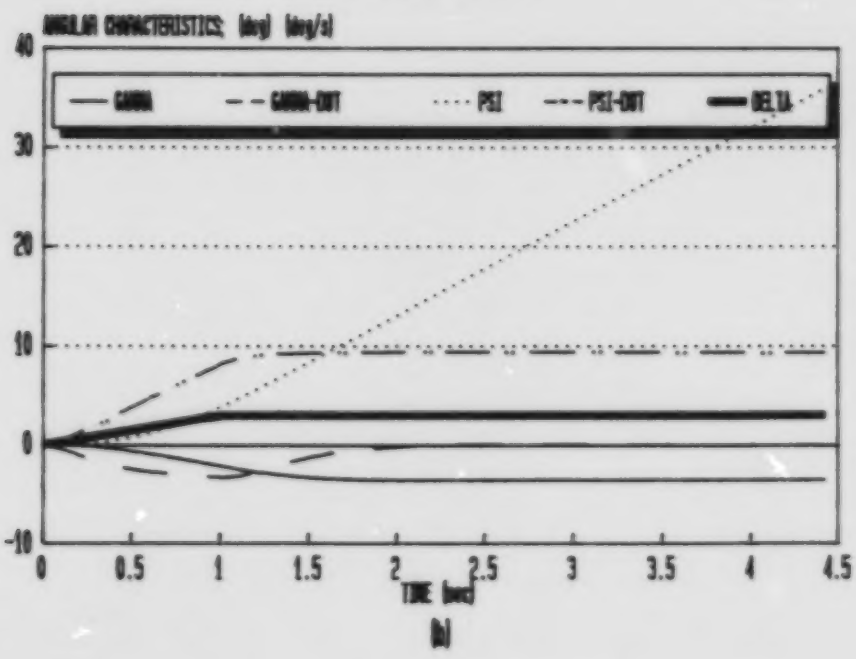
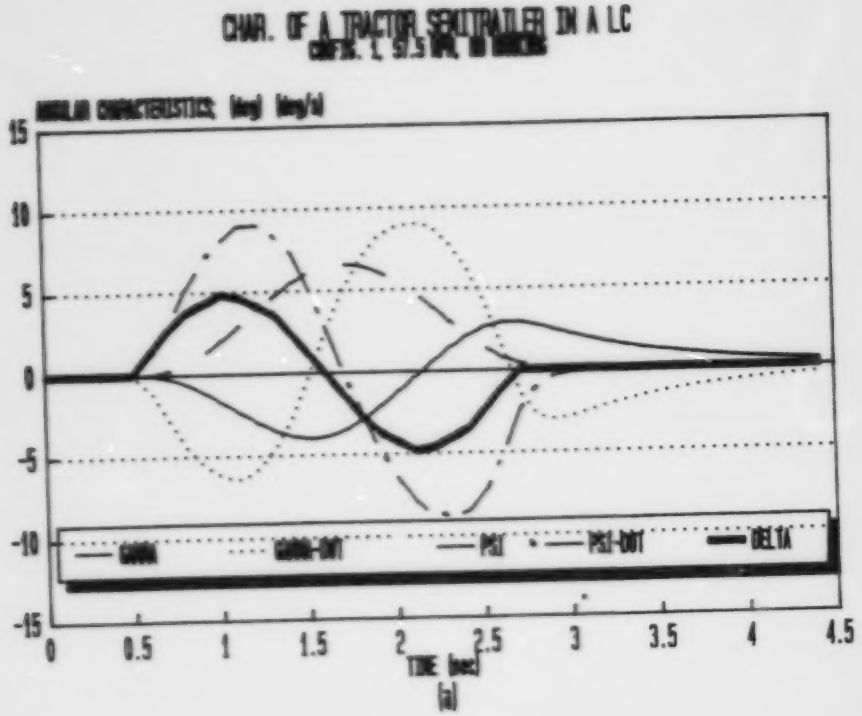


Figure 6 Steady State Characteristics in a Constant Radius Turn



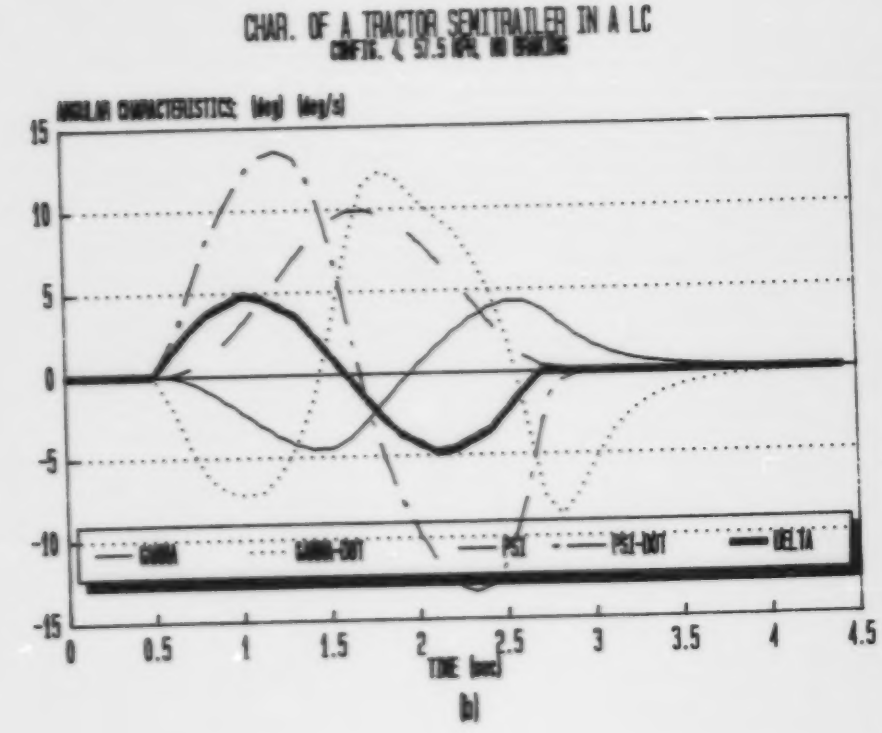


Figure 7 Characteristics of a Tractor Semitrailer in a Lane Change

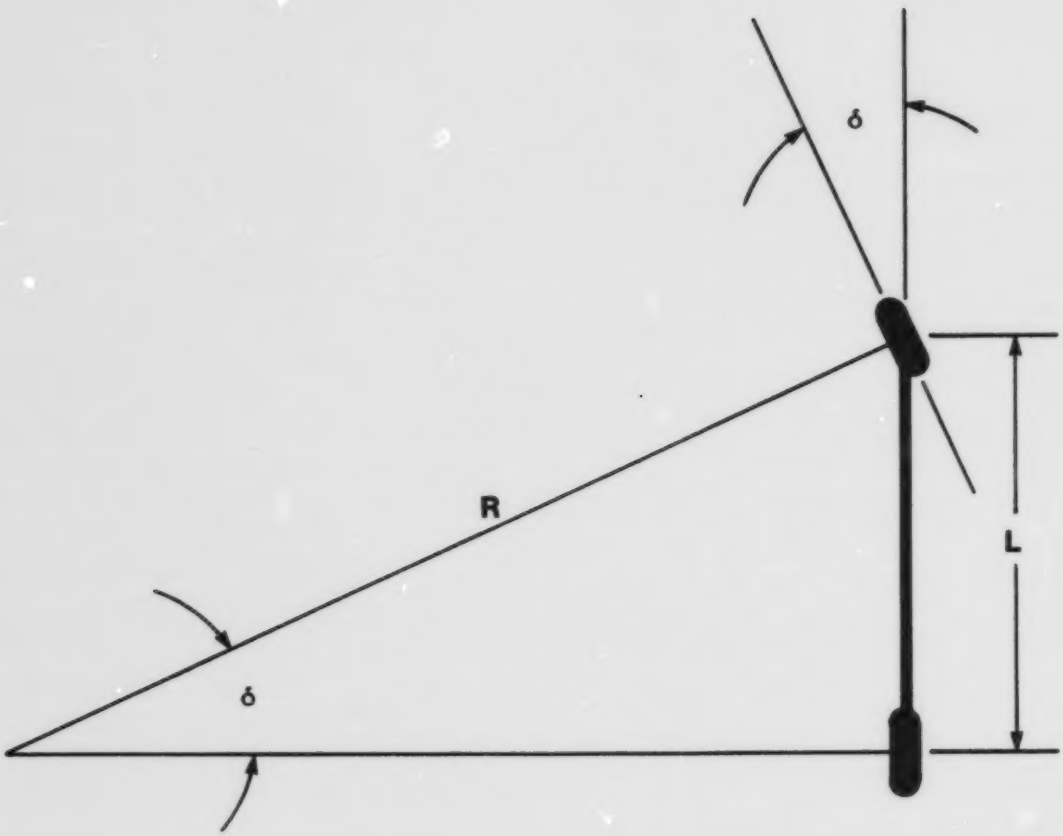


Figure 8 Geometric Approximation of the Turning Radius

CONFIGURATION : 4
SPEED : 57.5
MANEUVER : Constant Radius Turn
BRAKING : D1 - 20:40:40
DEVICE NAME : No Device
TORQUE EQN : -
CONSTANTS : -

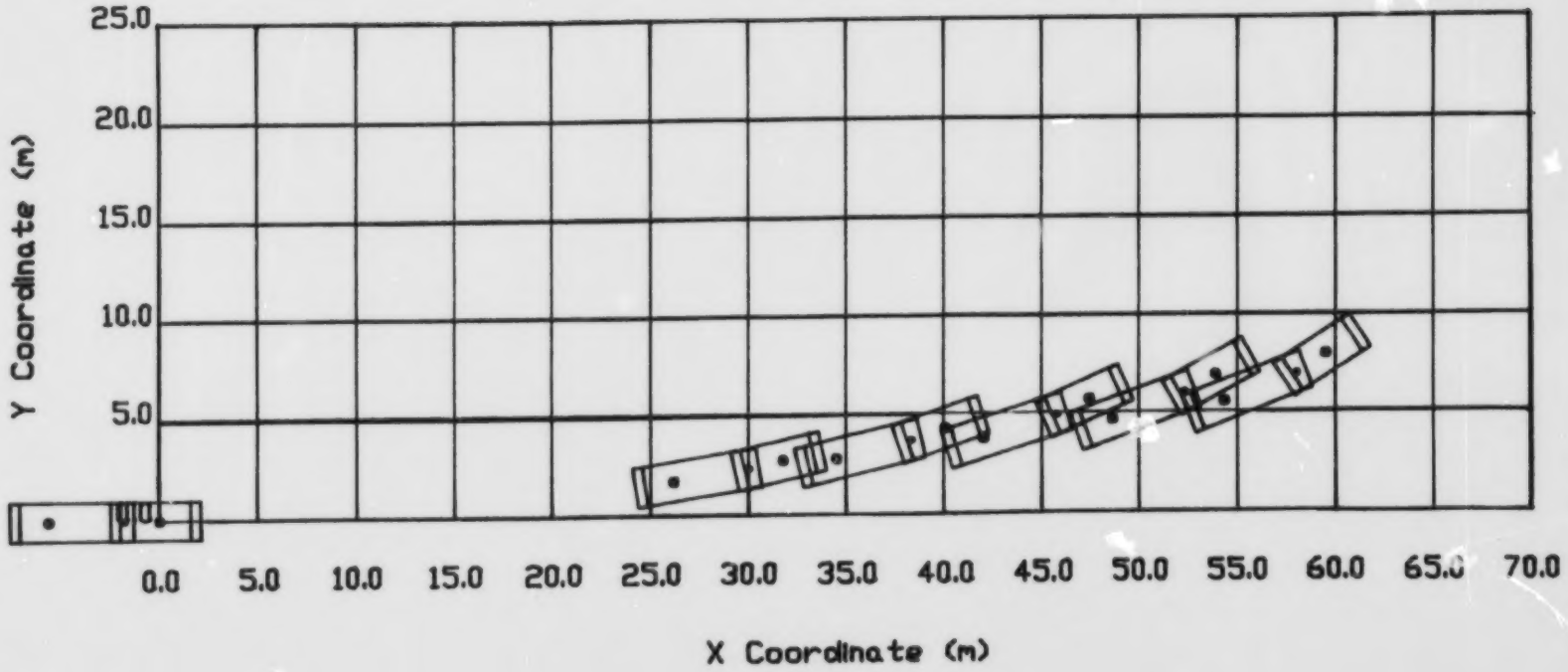


Figure 9 The D1 Braking Distribution (20:40:40)

CONFIGURATION : 4
SPEED : 57.5
MANEUVER : Constant Radius Turn
BRAKING : D2 - 33 : 33 : 33
DEVICE NAME : No Device
TORQUE EQN : -
CONSTANTS : -

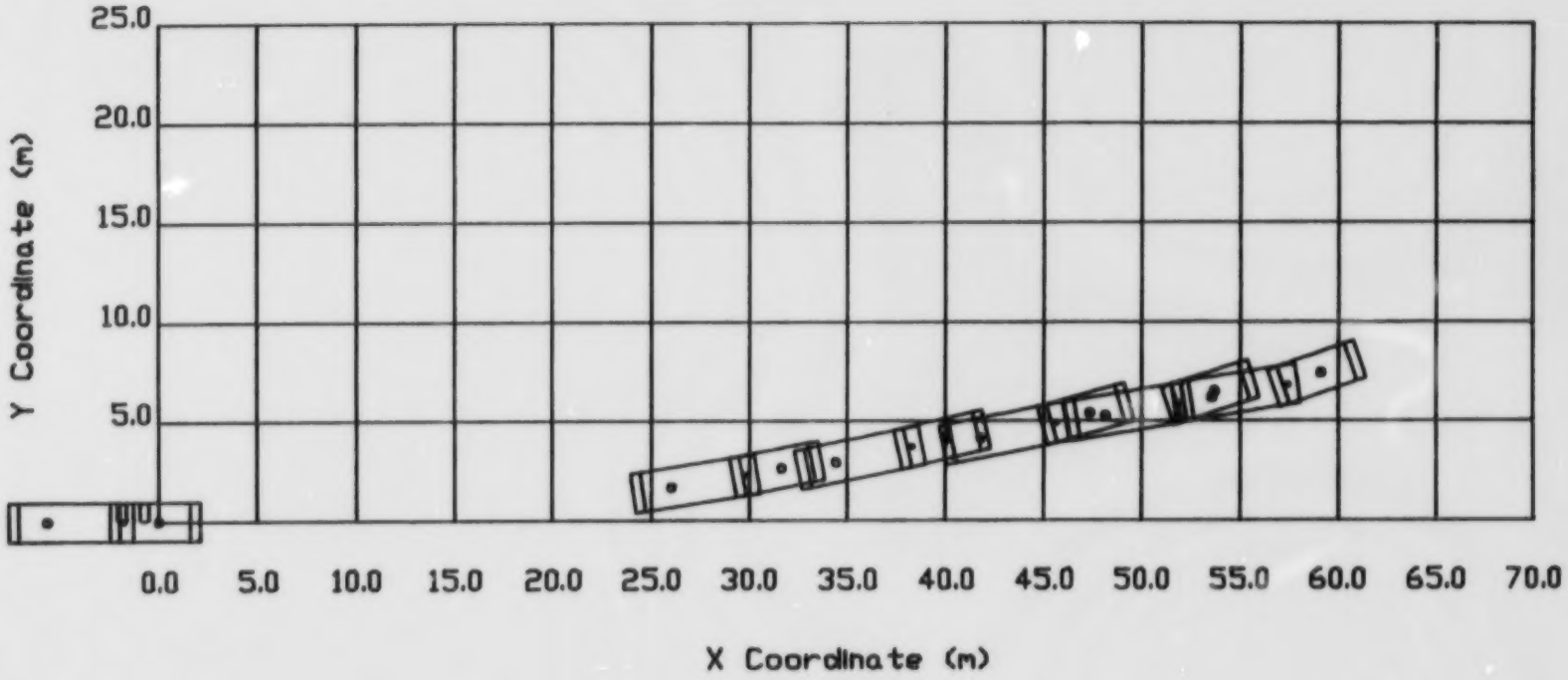


Figure 10 The D2 Braking Distribution (33:33:33)

CONFIGURATION : 4
SPEED : 57.5
MANEUVER : Constant Radius Turn
BRAKING : D3 - 27 : 36 : 37
DEVICE NAME : No Device
TORQUE EQN : -
CONSTANTS : -

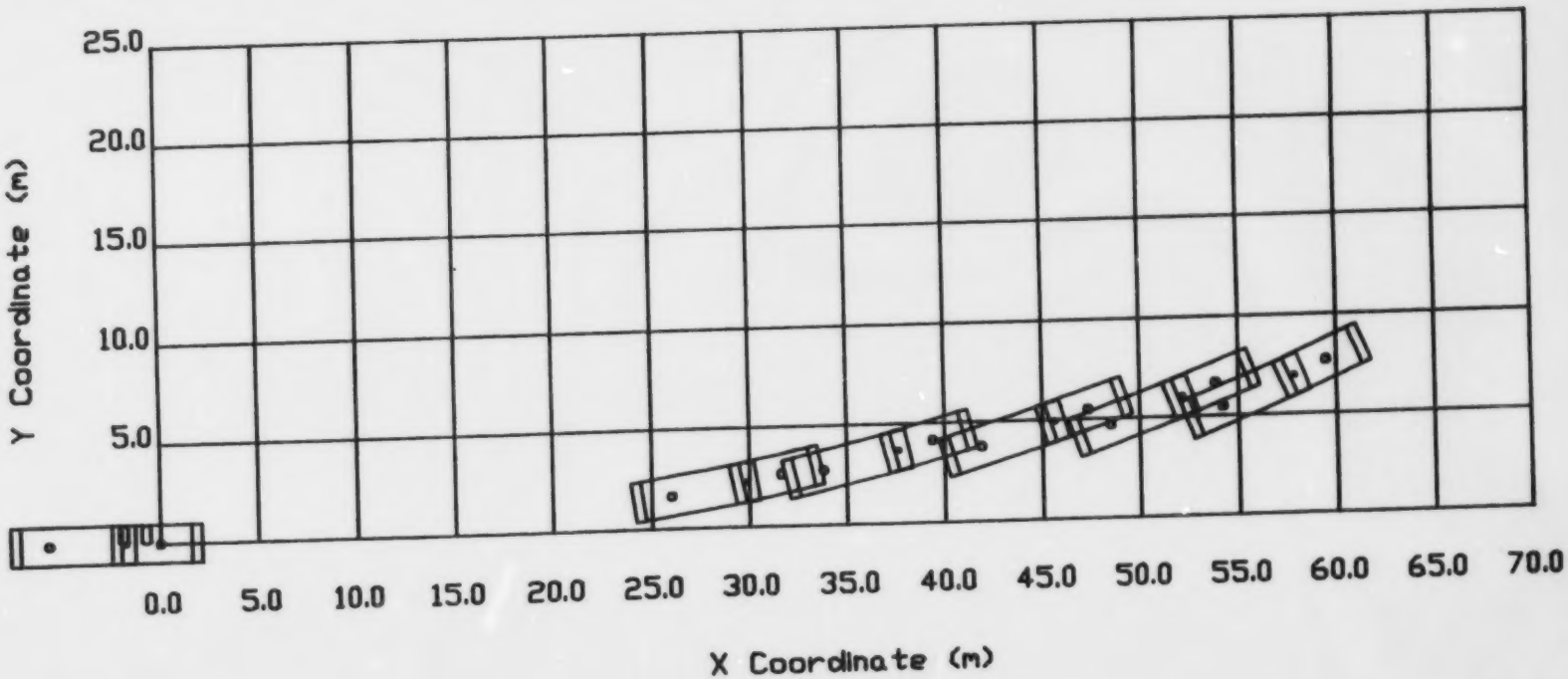


Figure 11 The D3 Braking Distribution (27:36:37)

CONFIGURATION : 4
SPEED : 57.5
MANEUVER : Constant Radius Turn
BRAKING : D4 - 15 : 45 : 40
DEVICE NAME : No Device
TORQUE EQN : -
CONSTANTS : -

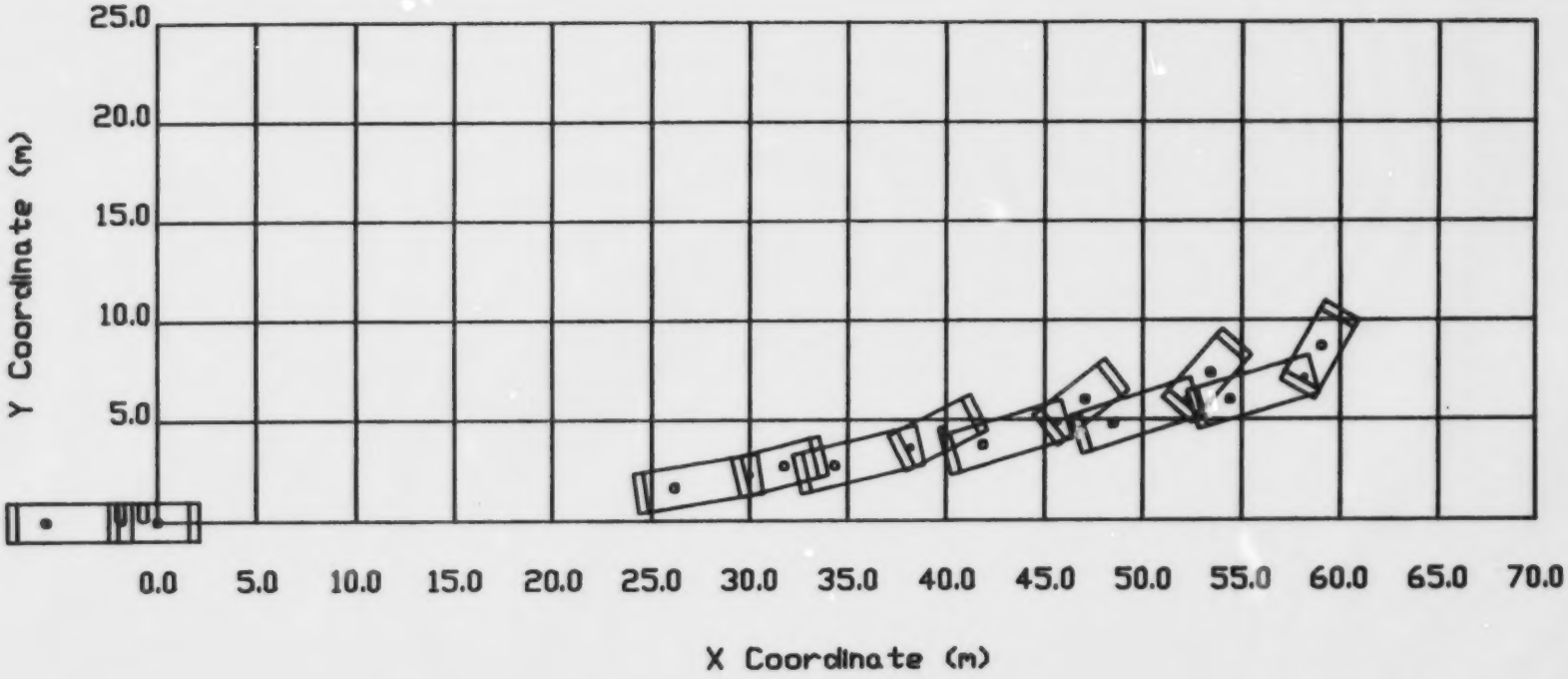


Figure 12 The D4 Braking Distribution (15:45:40)

CONFIGURATION : 4
SPEED : 57.5 km/h
MANEUVER : Constant Radius Turn
BRAKING : D5 - 15 : 50 : 35
DEVICE NAME : No Device
TORQUE EQN : -
CONSTANTS : -

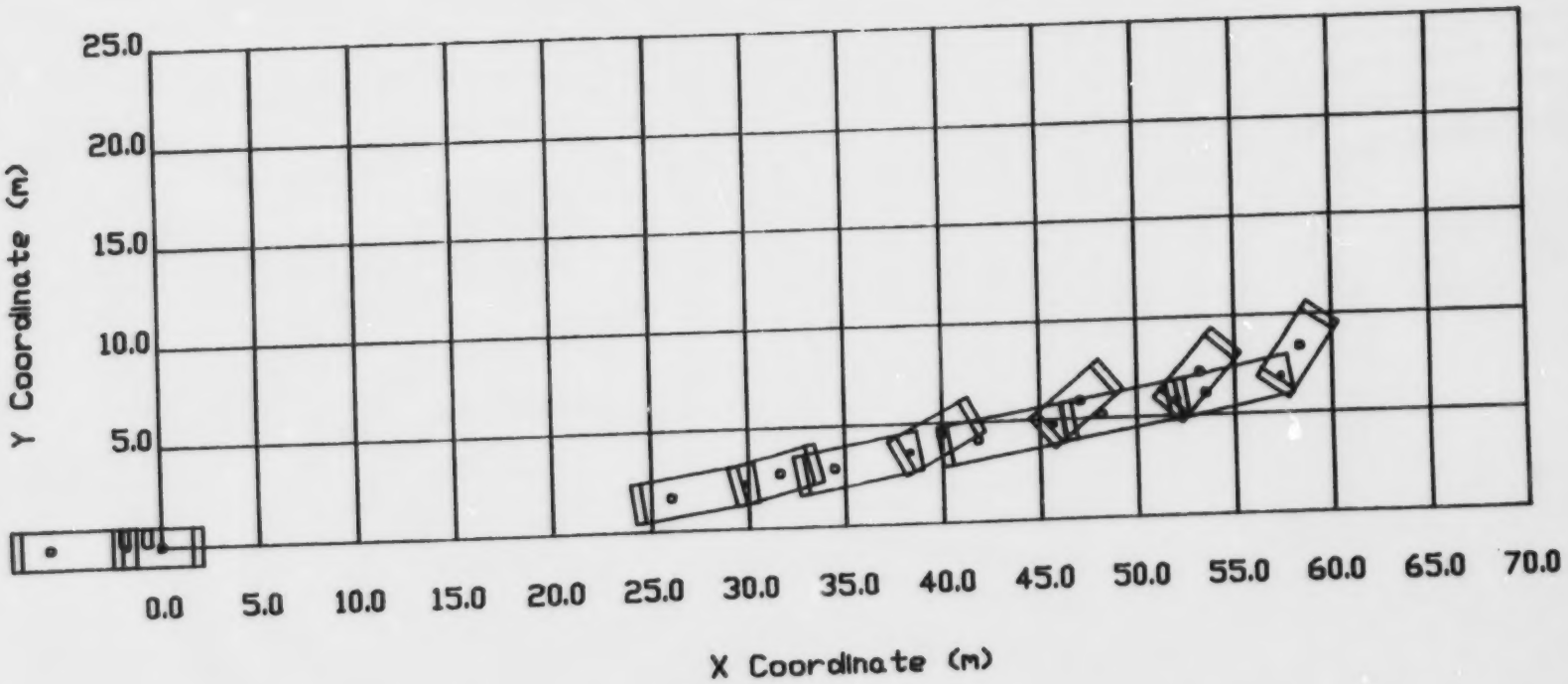
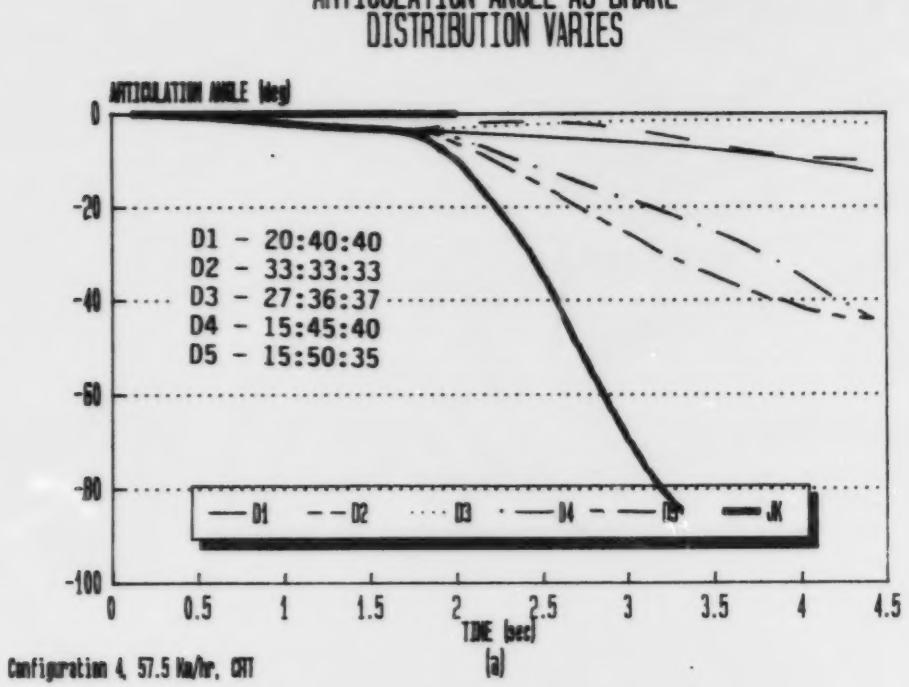


Figure 13 The D5 Braking Distribution (15:50:35)



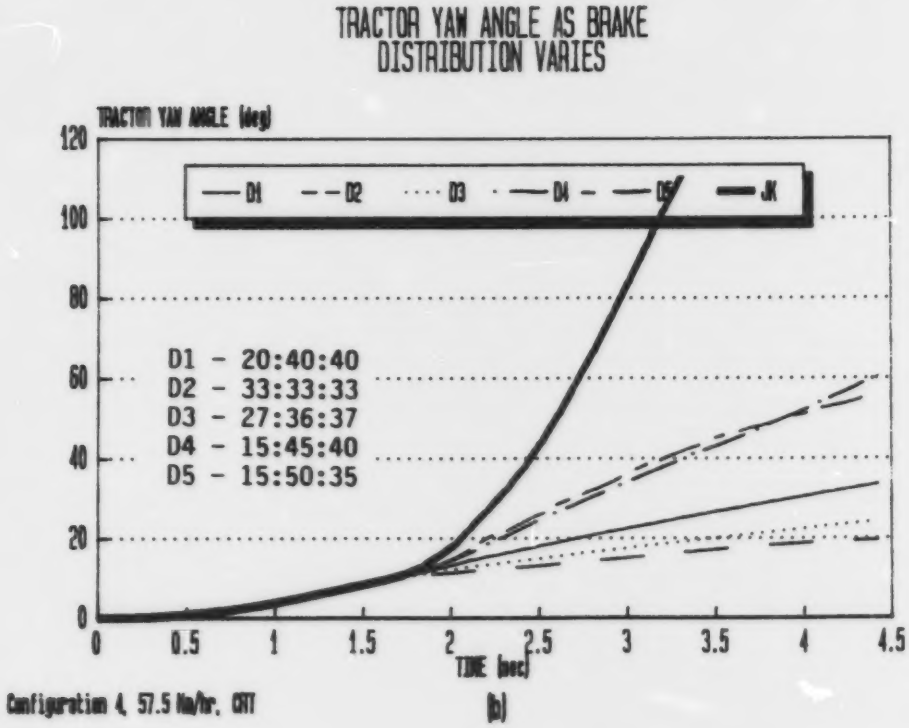
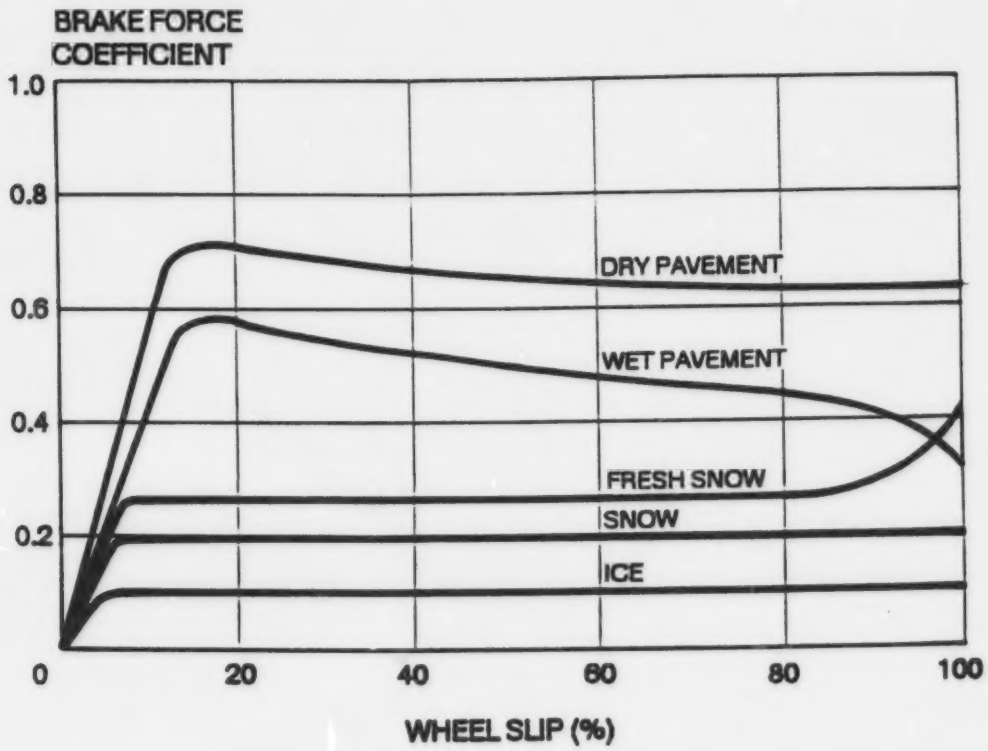


Figure 14 Brake Distribution Summary



Brake Force Coefficient vs. Wheel Slip

Figure 15 Brake Force Coefficient Versus Wheel Slip

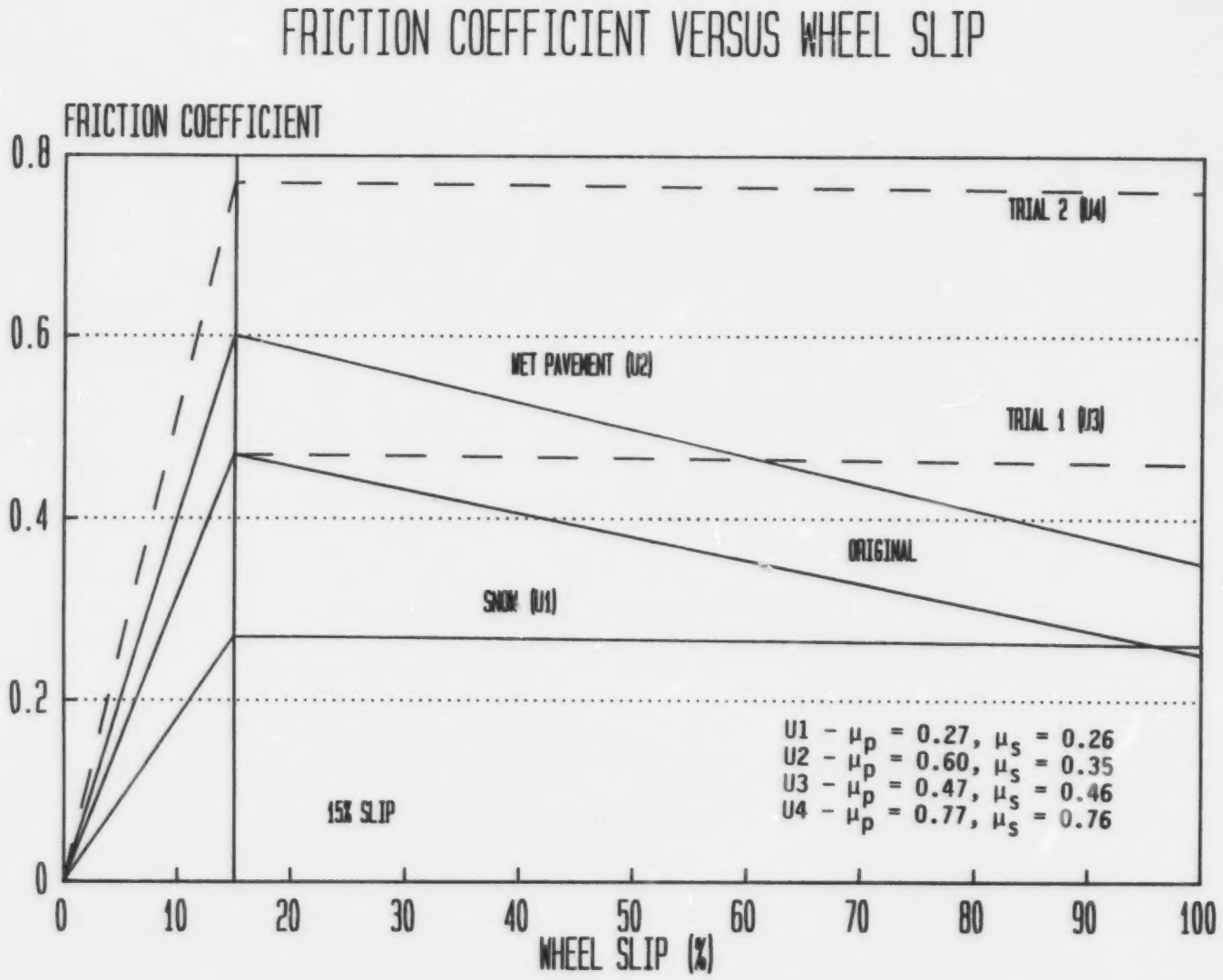


Figure 16 Simulated μ/slip Curves

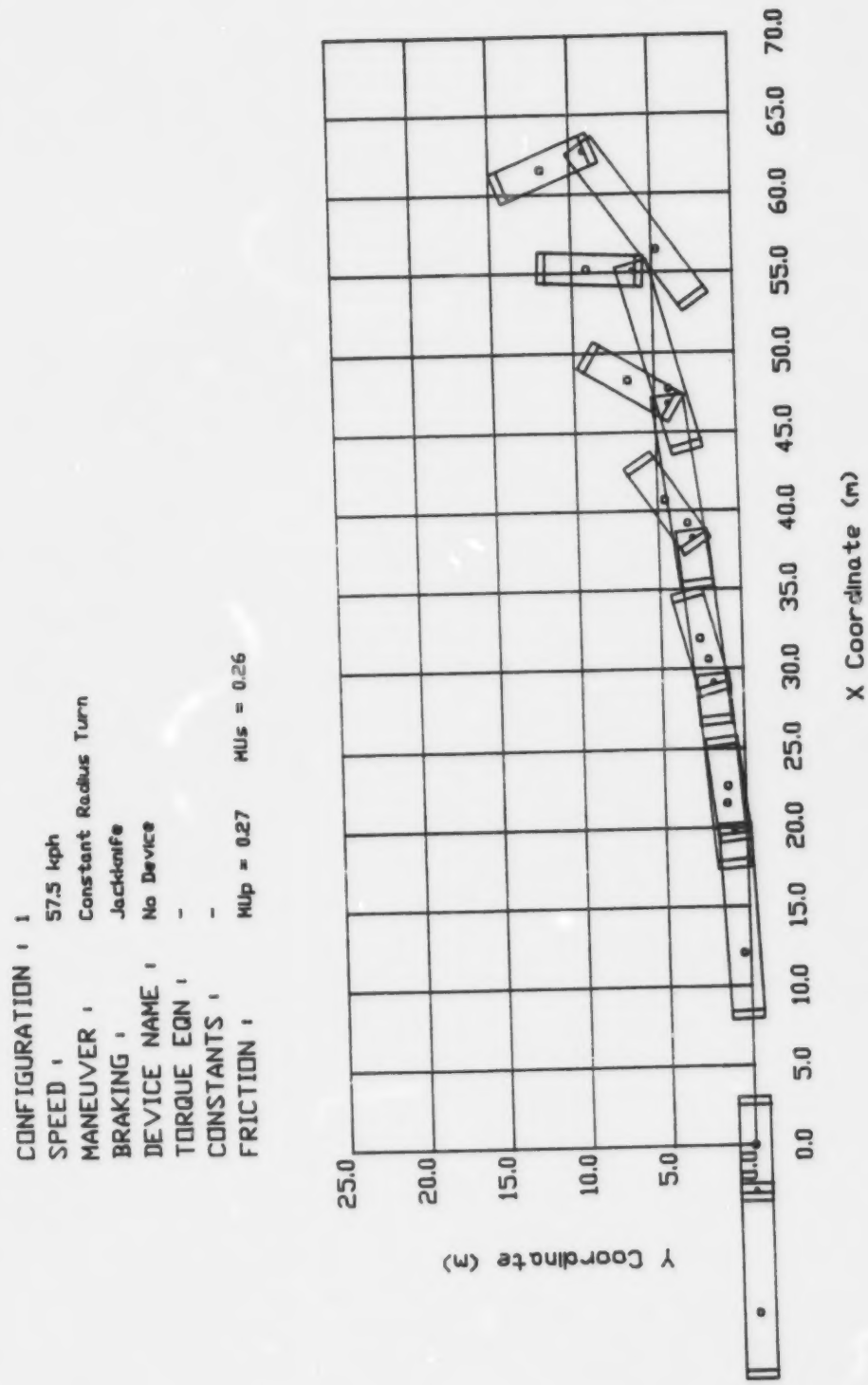


Figure 17 U1 (μ_p = 0.27, μ_s = 0.26) μ/slip Curve Simulation

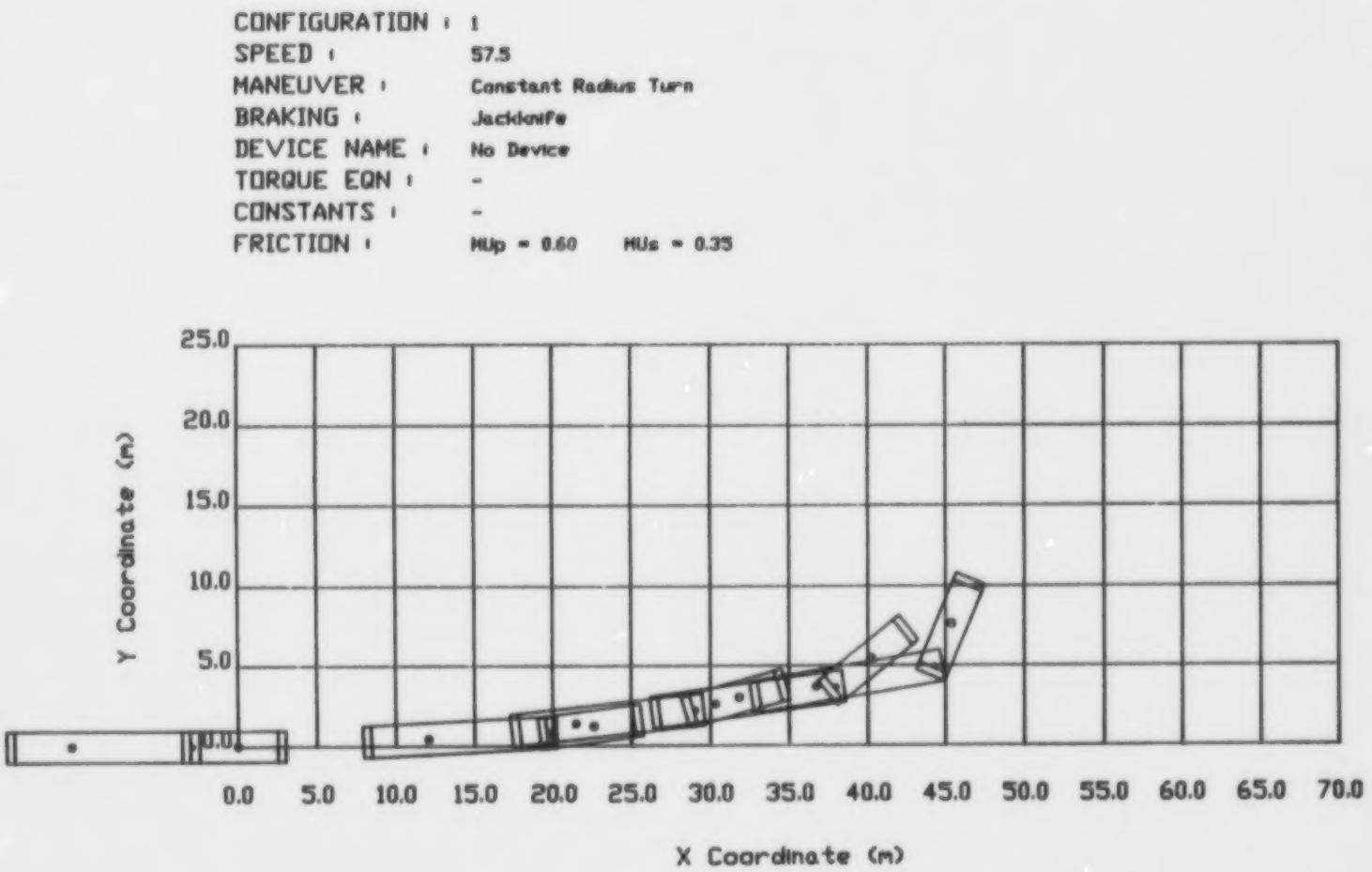


Figure 18 U2 (μ_p = 0.60, μ_s = 0.35) μ/slip Curve Simulation

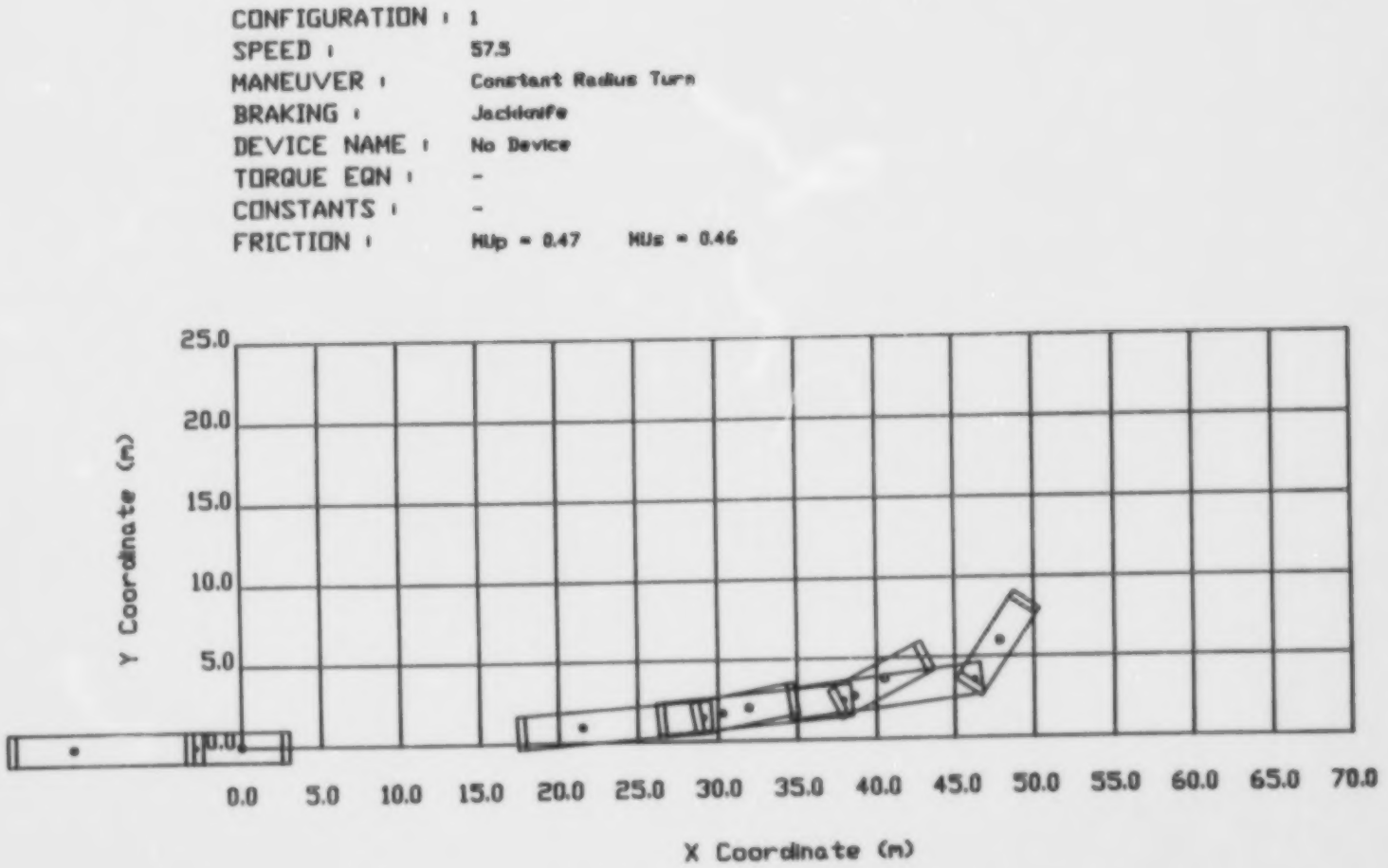


Figure 19 U3 (μ_p = 0.47, μ_s = 0.46) μ/slip Curve Simulation

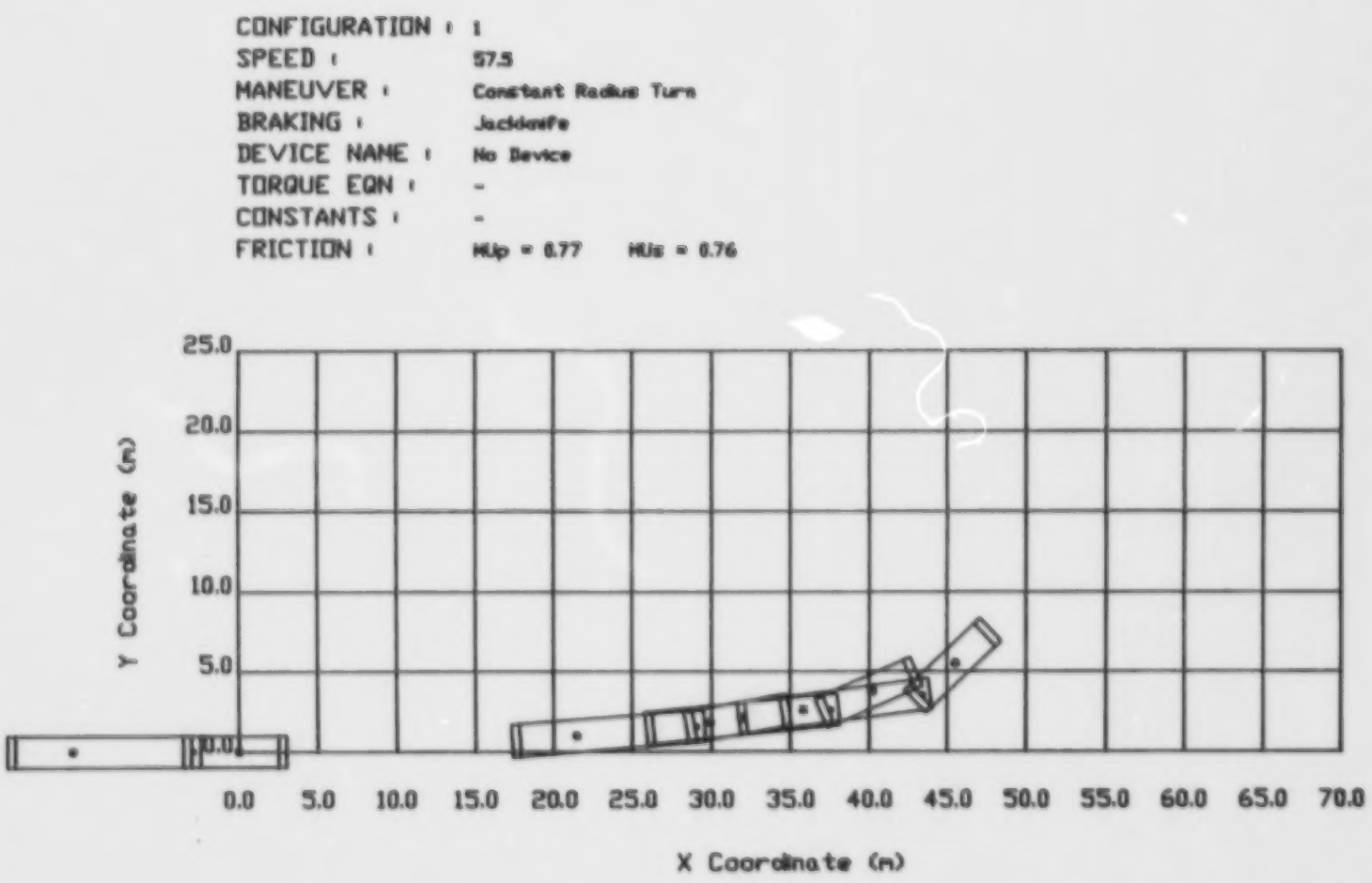
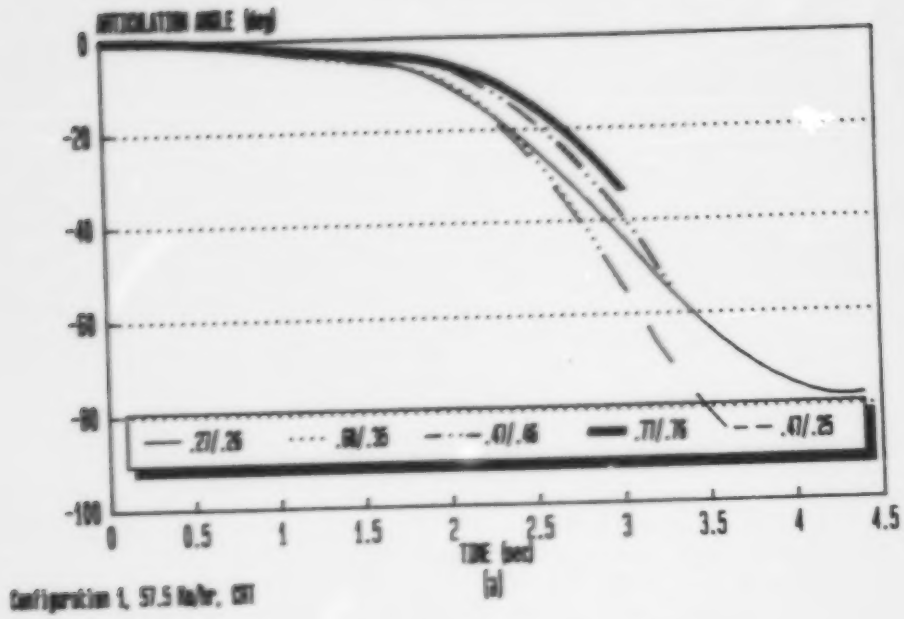


Figure 20 U4 (μ_p = 0.77, μ_s = 0.76) u/slip Curve Simulation



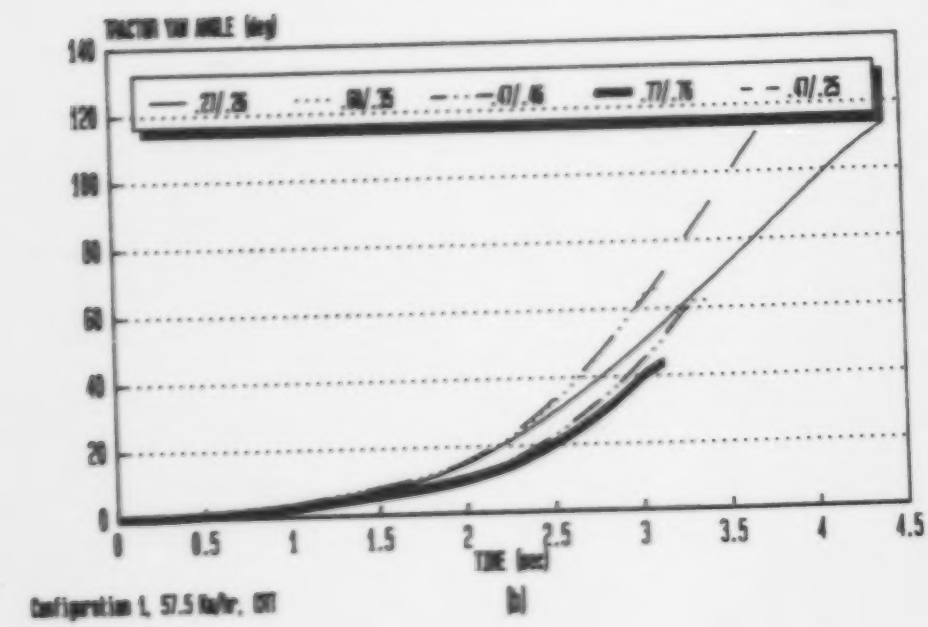


Figure 21 μ/slip Curve Simulation Summary

$$T = I \cdot \ddot{\Psi} + M \cdot a \cdot D$$

where, I = Tractor Moment of Inertia

$\ddot{\Psi}$ = Tractor Yaw Acceleration

M = Mass of Tractor

a = Tractor Acceleration

D = Moment Arm of 'ma' Vector

T = Restraining Torque

Figure 22 Active Anti-Jackknife Device Torque Calculation

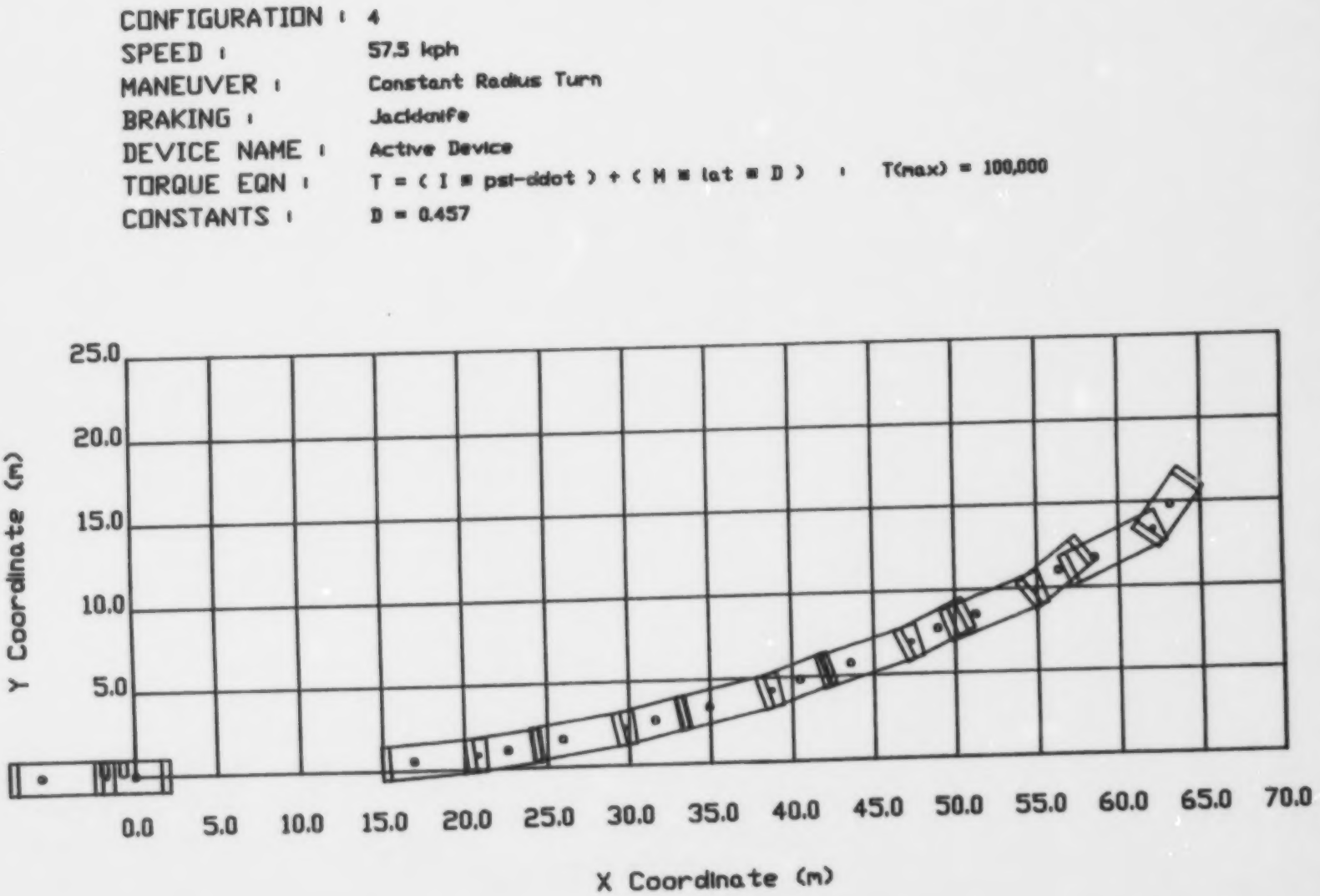
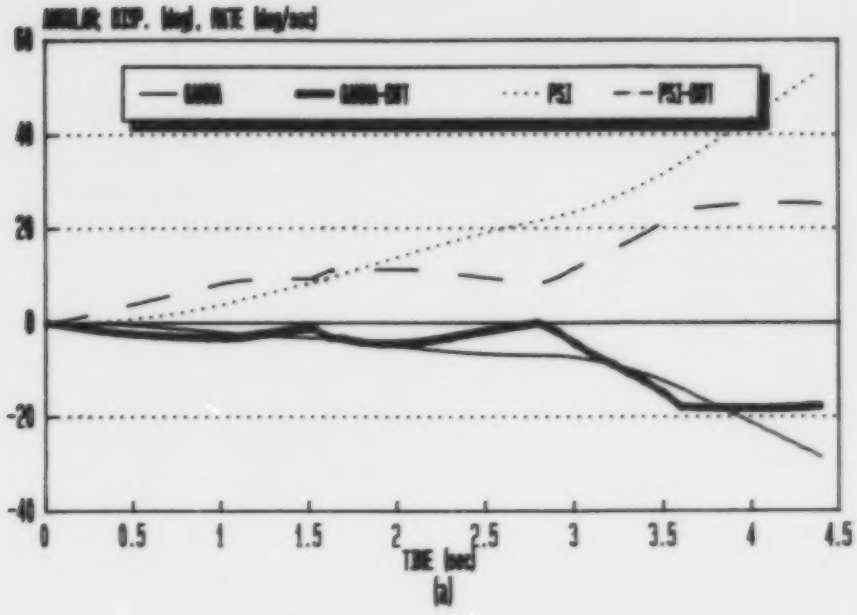


Figure 23 Active Anti-Jackknife Device Simulation Plot

JACKKNIFE CHARACTERISTICS WITH AN
ACTIVE ANTI-JACKKNIFE DEVICE



ACTIVE ANTI-JACKKNIFE DEVICE
APPLICATION TORQUE

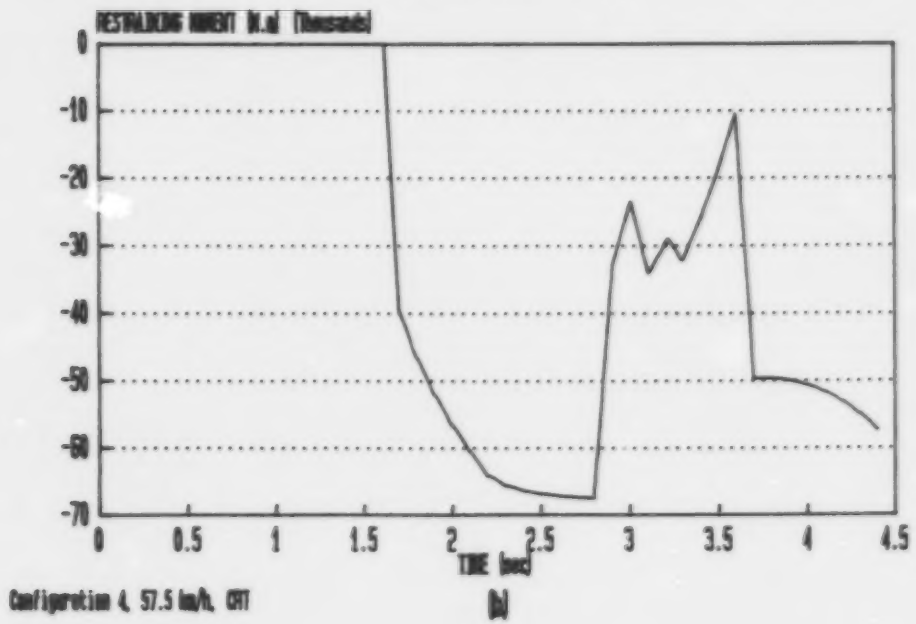
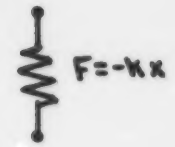


Figure 24 Active Anti-Jackknife Device Summary

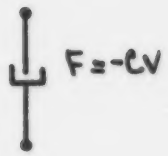
1. **Spring Type Device** $T=K\cdot\tan(\Gamma)\cdot D^2\cdot(\text{sign }\Gamma)$

$K_A = 50{,}000 \quad D_A = 0.46$
$K_B = 75{,}000 \quad D_B = 0.46$
$K_C = 100{,}000 \quad D_C = 0.46$



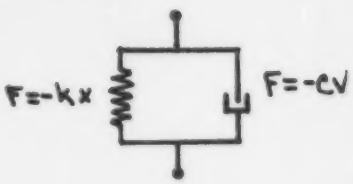
2. **Linear Damping Device (Dashpot)** $T=C\cdot\dot{\Gamma}\cdot D^2$

$C_A = 100{,}000 \quad D_A = 0.46$
$C_B = 150{,}000 \quad D_B = 0.46$
$C_C = 200{,}000 \quad D_C = 0.46$



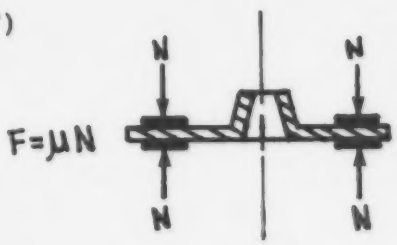
3. **Spring/Dashpot System** $T=[K\cdot\tan(\Gamma)\cdot D^2\cdot(\text{sign }\Gamma)]+[C\cdot\dot{\Gamma}\cdot D^2]$

$K_A = 75{,}000 \quad C_A = 75{,}000 \quad D_A = 0.46$
$K_B = 100{,}000 \quad C_B = 100{,}000 \quad D_B = 0.46$
$K_C = 50{,}000 \quad C_C = 50{,}000 \quad D_C = 0.46$
$K_D = 50{,}000 \quad C_D = 75{,}000 \quad D_D = 0.46$
$K_E = 50{,}000 \quad C_E = 100{,}000 \quad D_E = 0.46$
$K_F = 75{,}000 \quad C_F = 50{,}000 \quad D_F = 0.46$



4. **Constant Damping Device** $T=N\cdot\mu\cdot D\cdot(\text{sign }\dot{\Gamma})$

$N_A = 20{,}000 \quad \mu_A = 0.50 \quad D_A = 0.46$
$N_B = 40{,}000 \quad \mu_B = 0.50 \quad D_B = 0.46$
$N_C = 80{,}000 \quad \mu_C = 0.50 \quad D_C = 0.46$
$N_D = 120{,}000 \quad \mu_D = 0.50 \quad D_D = 0.46$
$N_E = 160{,}000 \quad \mu_E = 0.50 \quad D_E = 0.46$



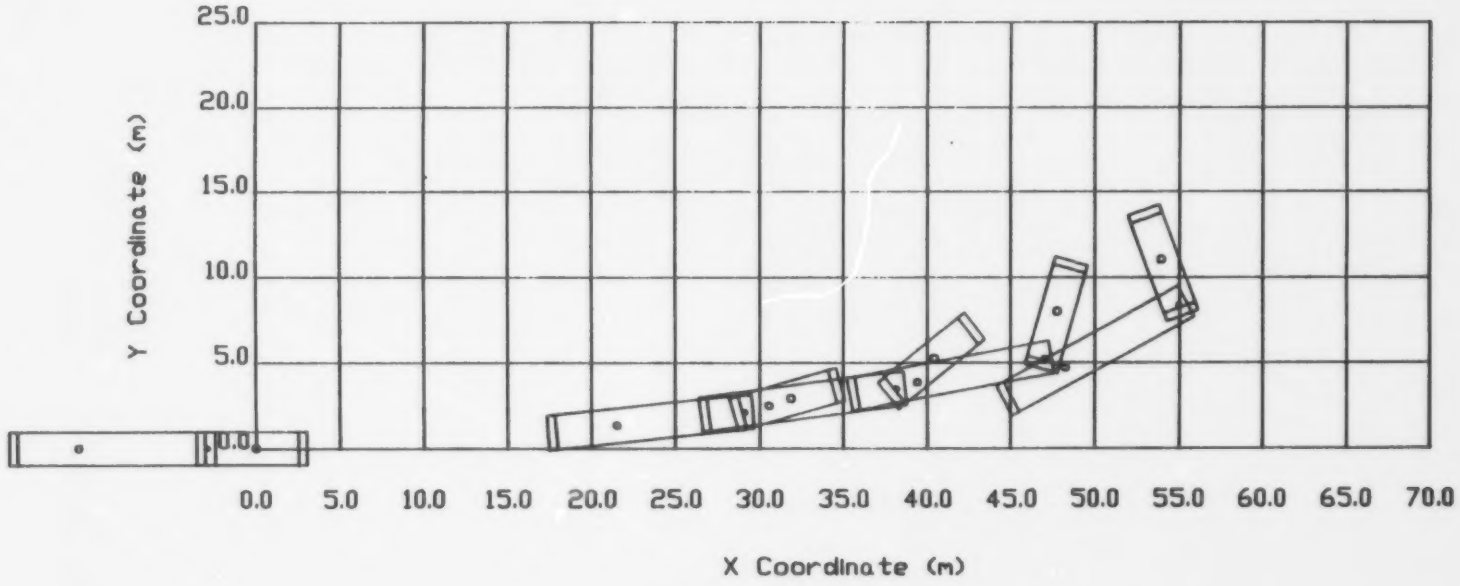
5. **Variable Damping Device** $T=(K1\cdot\dot{\Gamma}^2)\cdot(K2\cdot\Gamma)\cdot(\text{sign }\dot{\Gamma})$

$K1_A = 100{,}000 \quad K2_A = 10$
$K1_B = 200{,}000 \quad K2_B = 10$
$K1_C = 300{,}000 \quad K2_C = 10$
$K1_D = 400{,}000 \quad K2_D = 10$
$K1_E = 500{,}000 \quad K2_E = 10$
$K1_F = 700{,}000 \quad K2_F = 10$
$K1_G = 900{,}000 \quad K2_G = 10$
$K1_H = 500{,}000 \quad K2_H = 15$

Figure 25 Summary of the Passive Anti-Jackknife Devices Studied

Figure 26 Spring Type Device (k = 50,000 N/m)

CONFIGURATION : 1
SPEED : 57.5 km/h
MANEUVER : Constant Radius Turn
BRAKING : Jackknife
DEVICE NAME : Spring Type
TORQUE EQN : T = K * tan(gam) * D**2 * sign(gam)
CONSTANTS : K = 50,000 D = 0.457



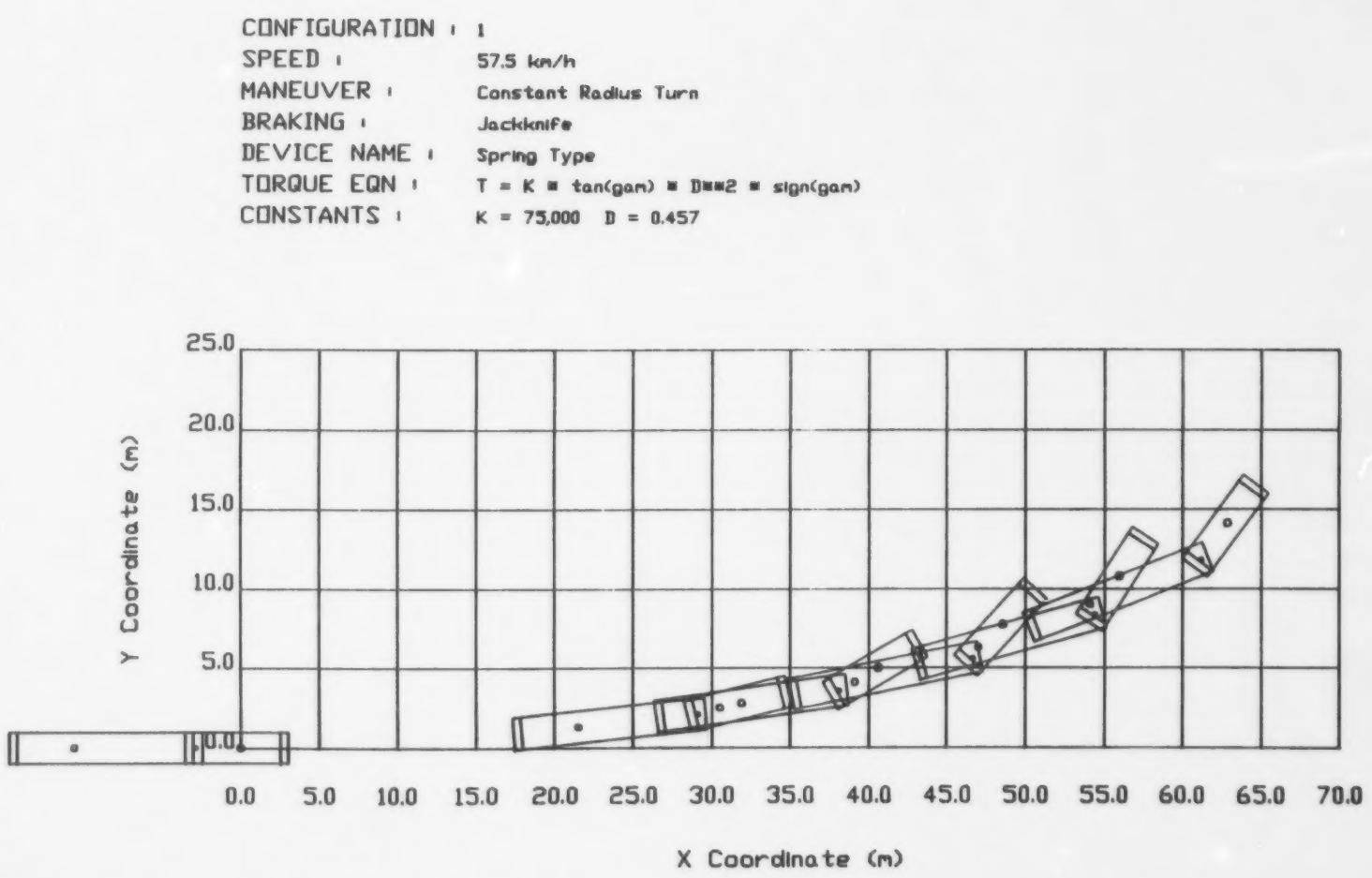


Figure 27 Spring Type Device (k = 75,000 N/m)

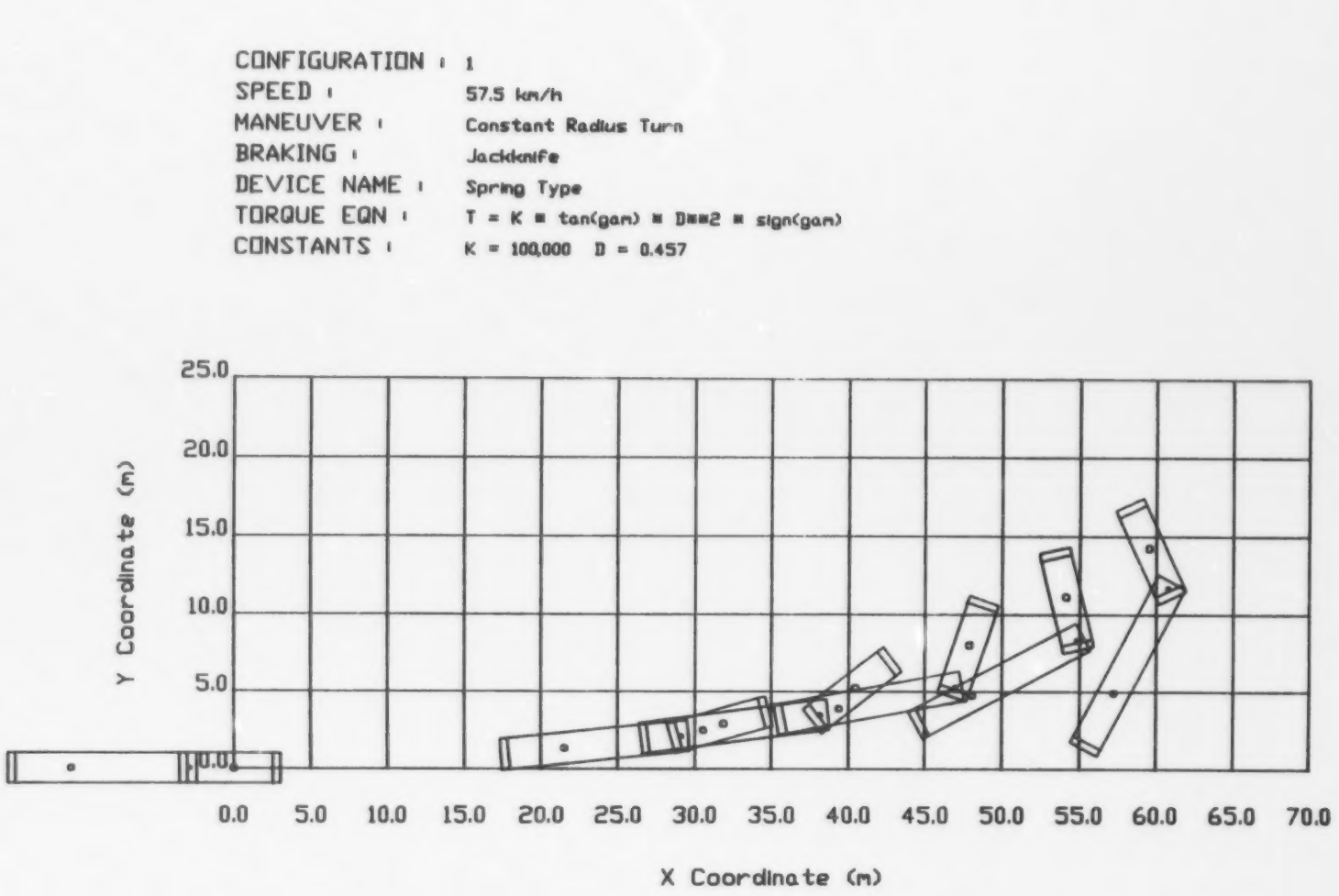

CONFIGURATION : 1
SPEED : 57.5 km/h
MANEUVER : Constant Radius Turn
BRAKING : Jackknife
DEVICE NAME : Spring Type
TORQUE EQN : T = K * tan(gam) * D**2 * sign(gam)
CONSTANTS : K = 100,000 D = 0.457
25.0
20.0
15.0
10.0
5.0
0.0
Y Coordinate (m)
0.0 5.0 10.0 15.0 20.0 25.0 30.0 35.0 40.0 45.0 50.0 55.0 60.0 65.0 70.0
X Coordinate (m)

Figure 28 Spring Type Device (K = 100,000 N.m

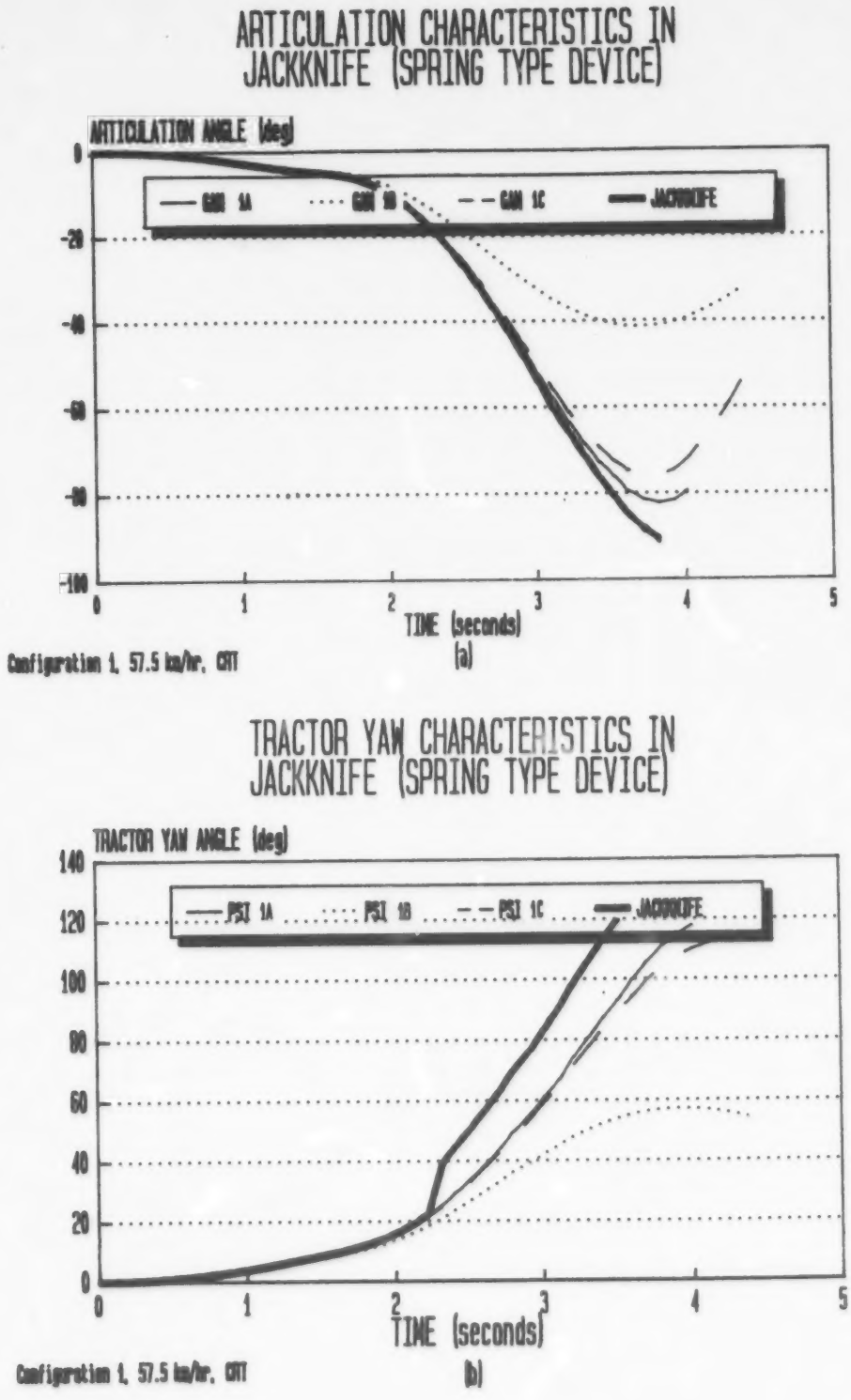


Figure 29 Jackknife Characteristics with a Spring Type Device

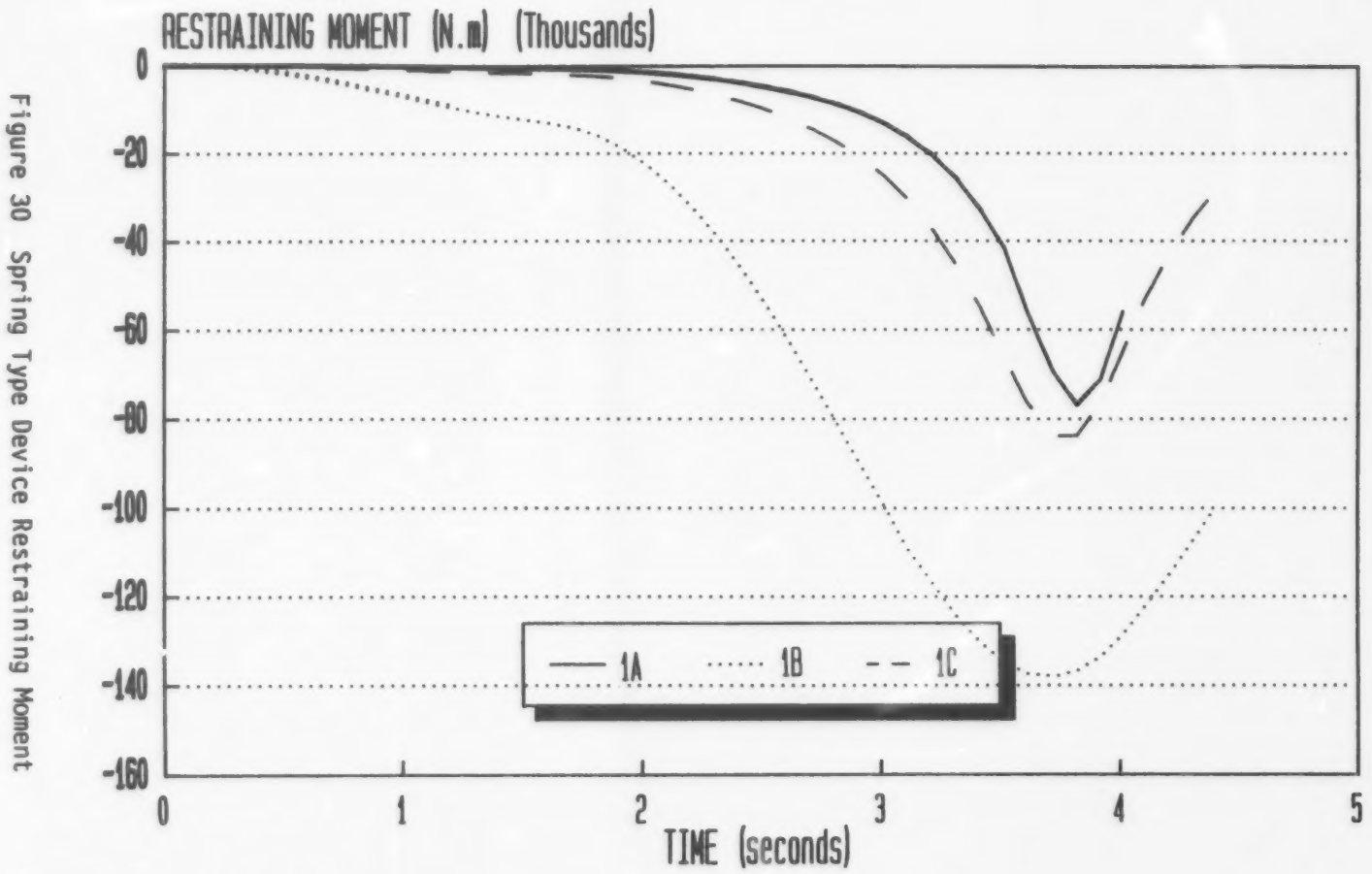


Figure 30 Spring Type Device Restraining Moment

CONFIGURATION : 4
SPEED : 57.5 km/h
MANEUVER : Constant Radius Turn
BRAKING : Jackknife
DEVICE NAME : Linear Damping (Dashpot)
TORQUE EQN : T = C * gamdot * D**2
CONSTANTS : C = 200,000 D = 0.457

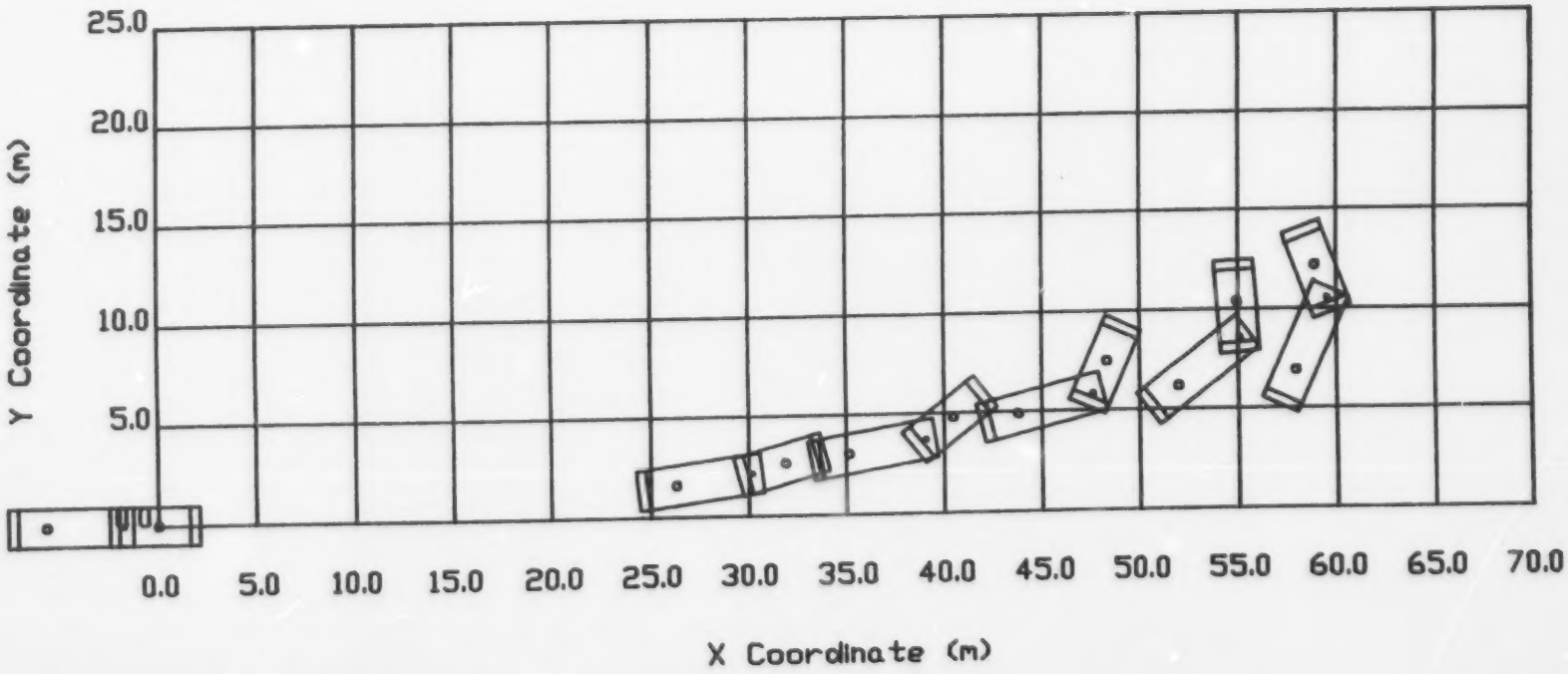
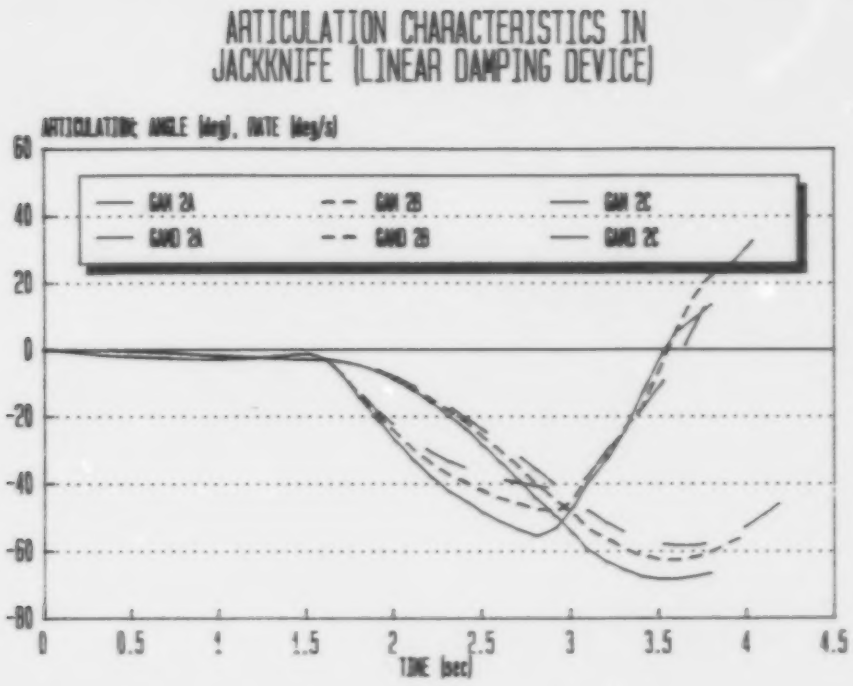


Figure 31 Linear Damping Device (C = 200,000 N•s/m)



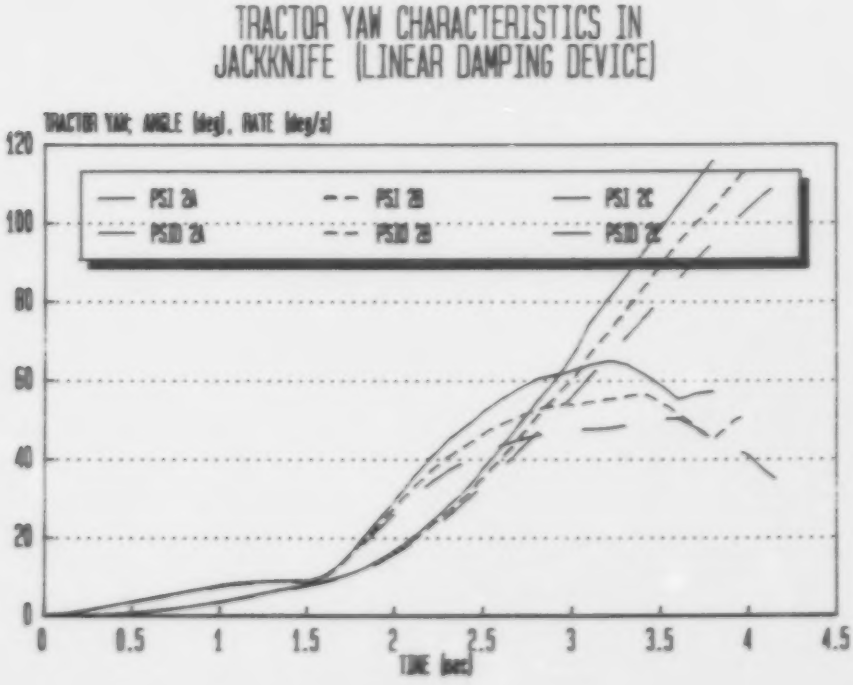


Figure 32 Jackknife Characteristics with a Linear Damping Device

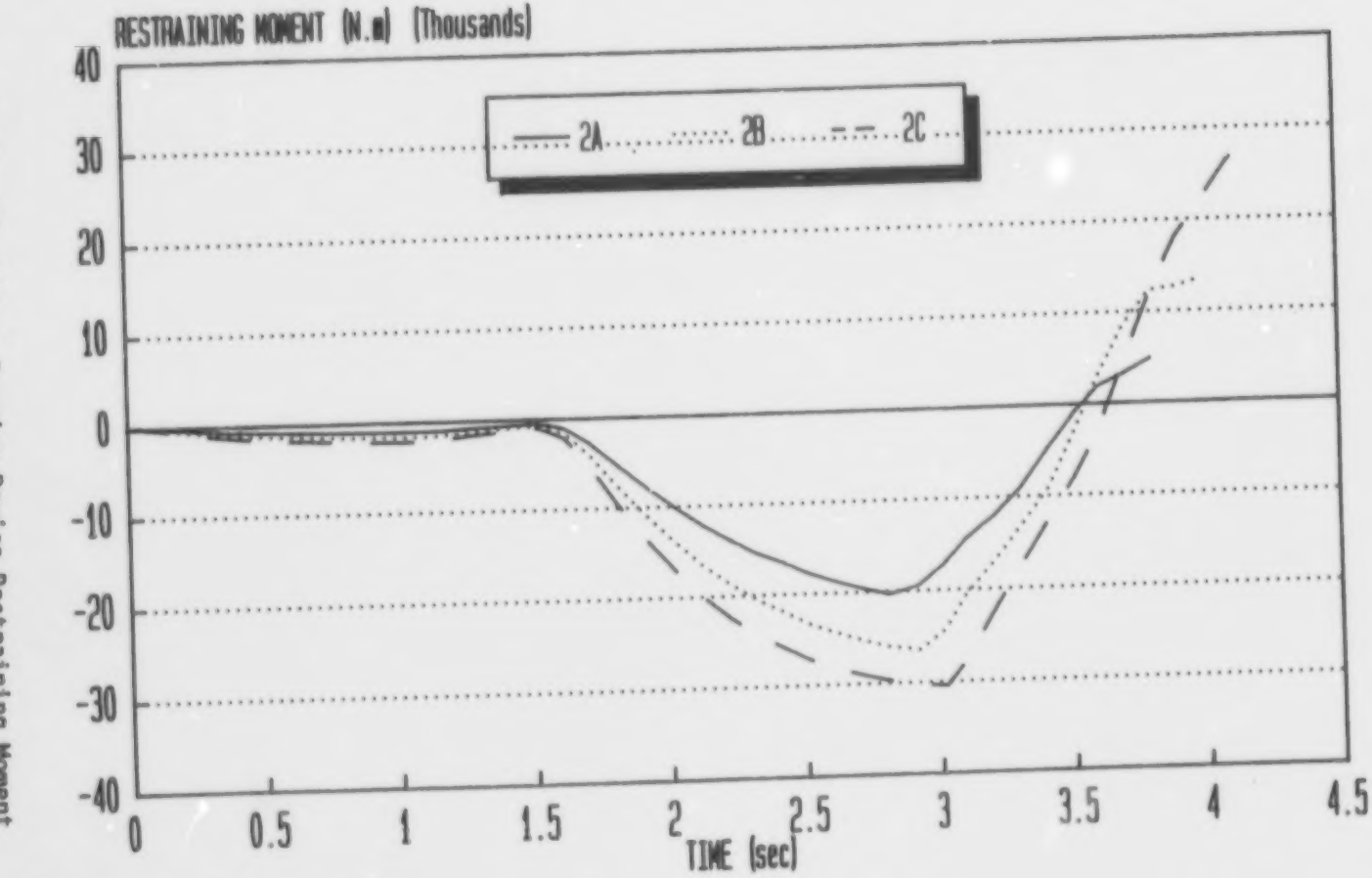


Figure 33 Linear Damping Device Restraining Moment

Configuration 4, 57.5 km/h, CRT

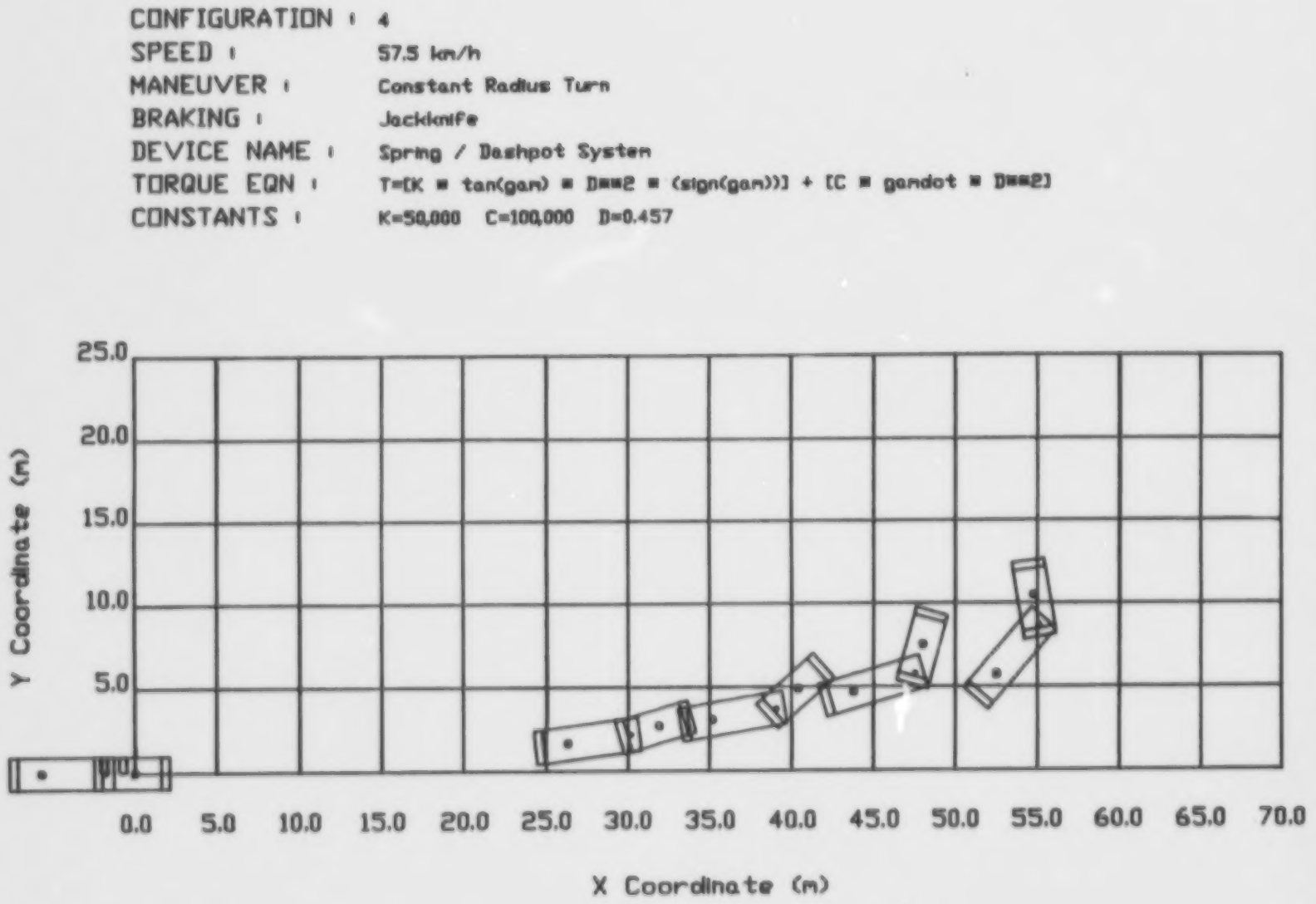
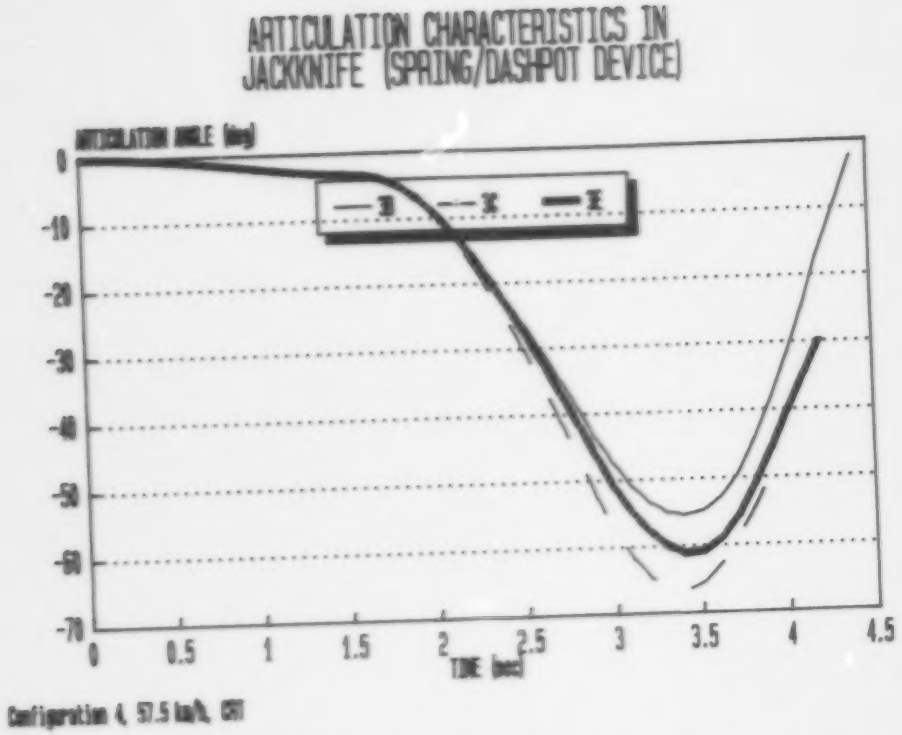


Figure 34 Spring/Dashpot System (K = 50,000, C = 100,000)



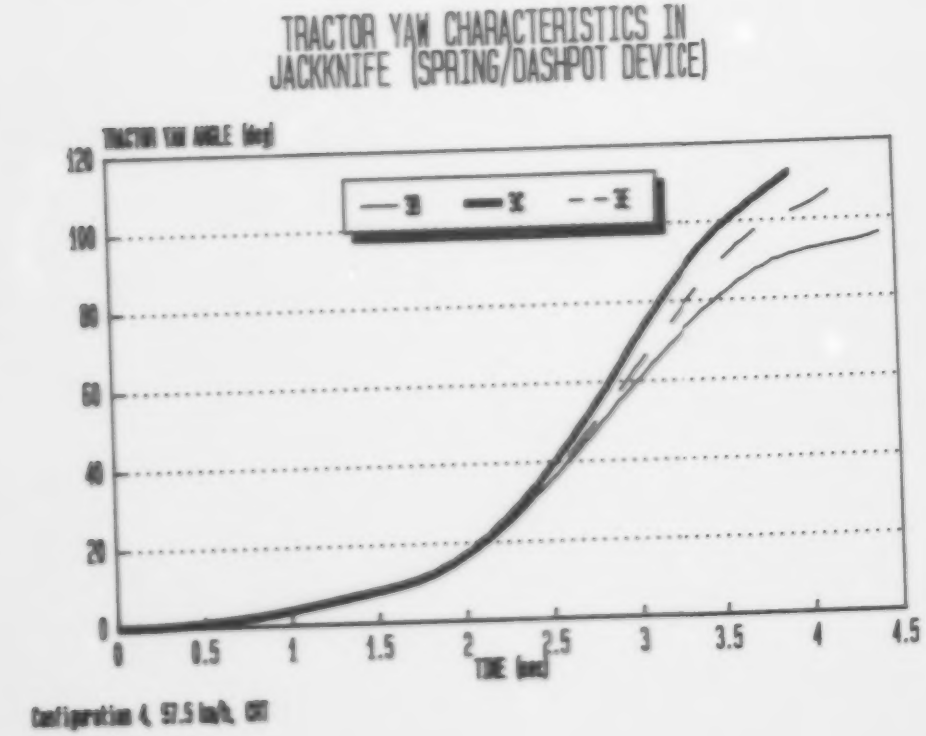


Figure 35 Jackknife Characteristics with a Spring/Dashpot System

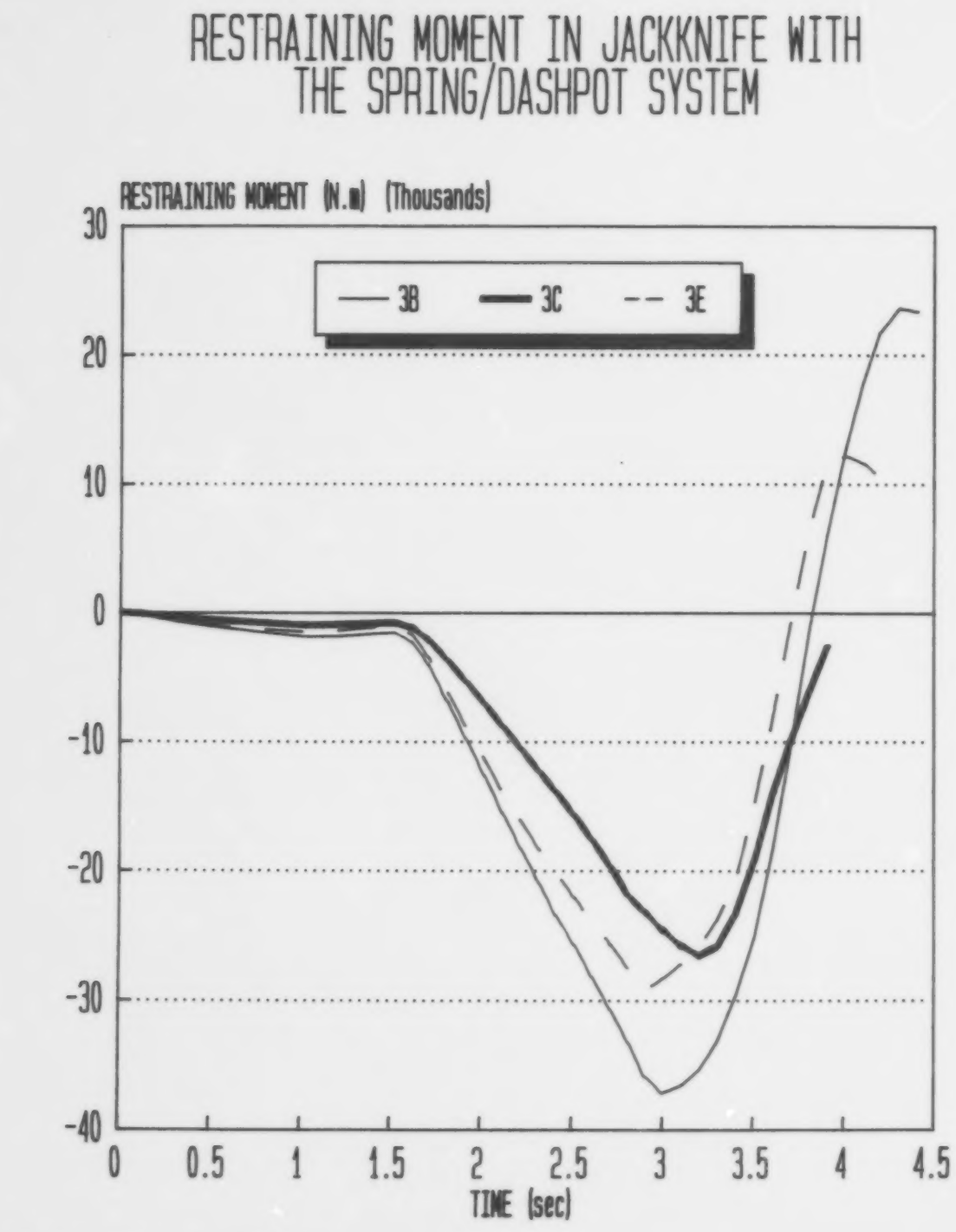


Figure 36 Spring/Dashpot System Restraining Moment

CONFIGURATION : 4
SPEED : 57.5 km/h
MANEUVER : Constant Radius Turn
BRAKING : Jackknife
DEVICE NAME : Constant Damping
TORQUE EQN : T = K * 0.5 * sign(gandot)
CONSTANTS : K = 80,000

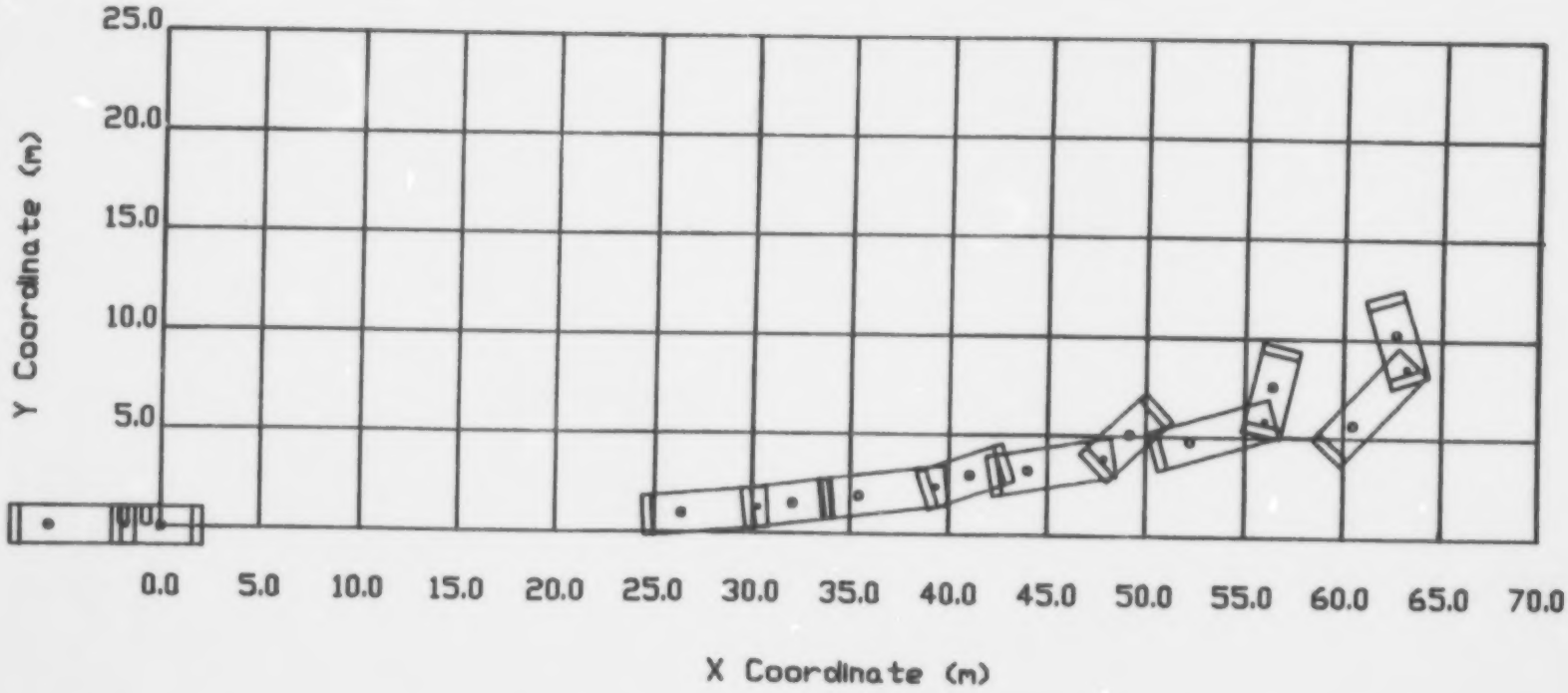


Figure 37 Constant Damping Device (N = 80,000 N)

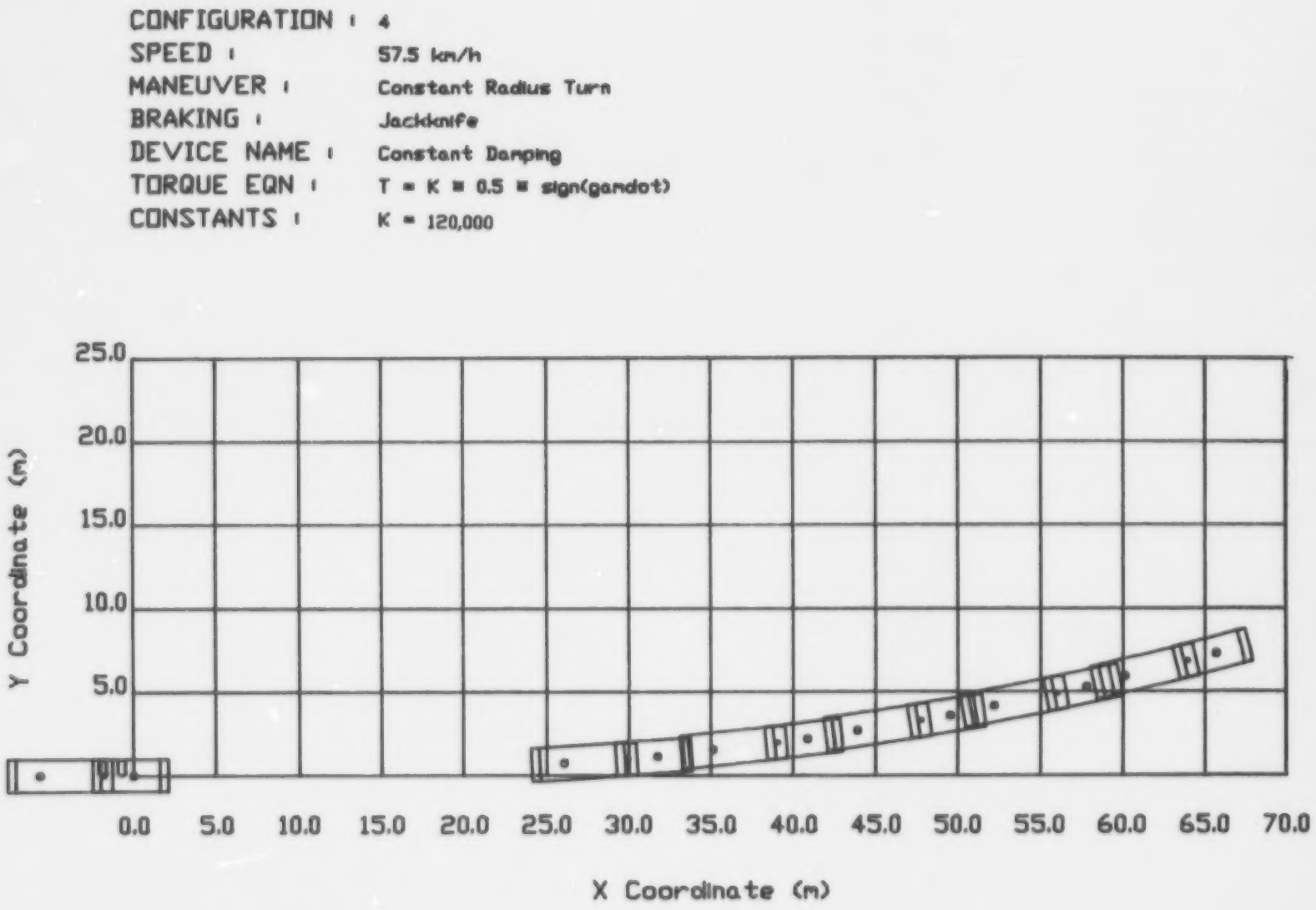
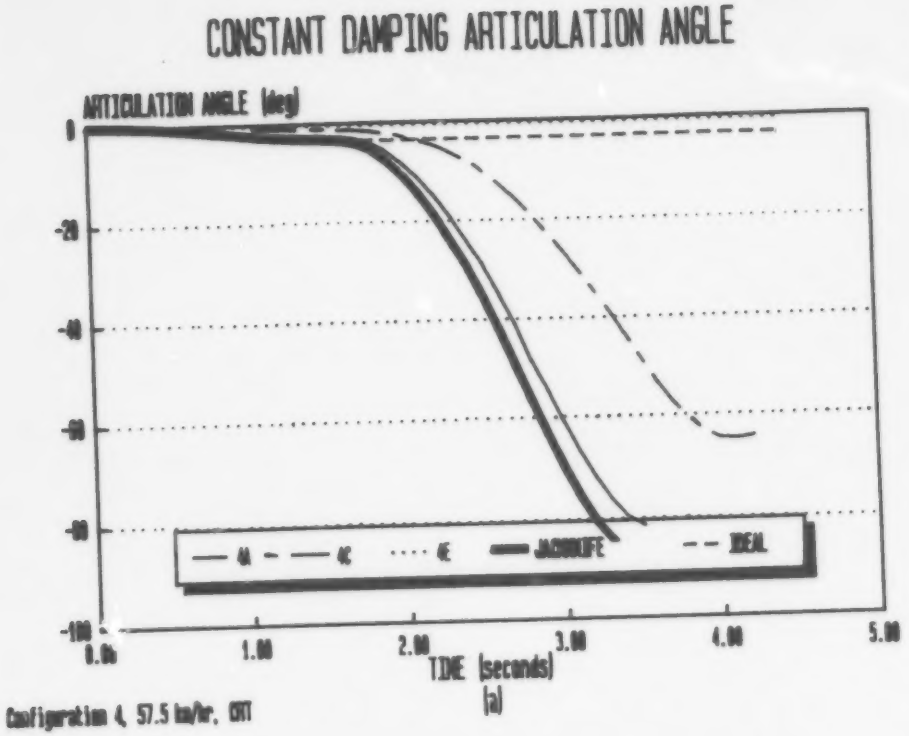


Figure 38 Constant Damping Device (N = 120,000 N)



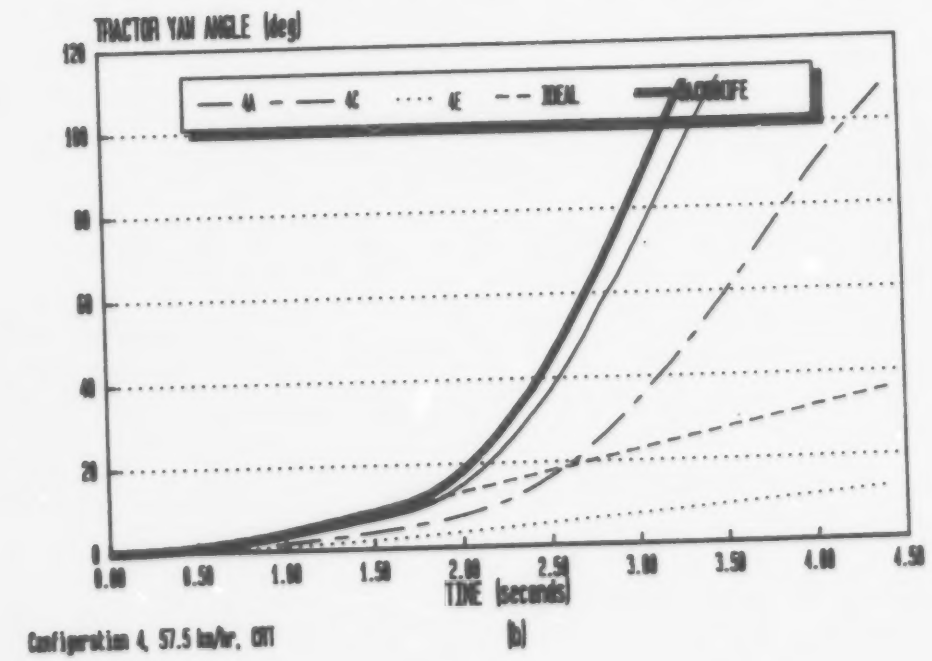


Figure 39 Jackknife Characteristics with a Constant Damping Device

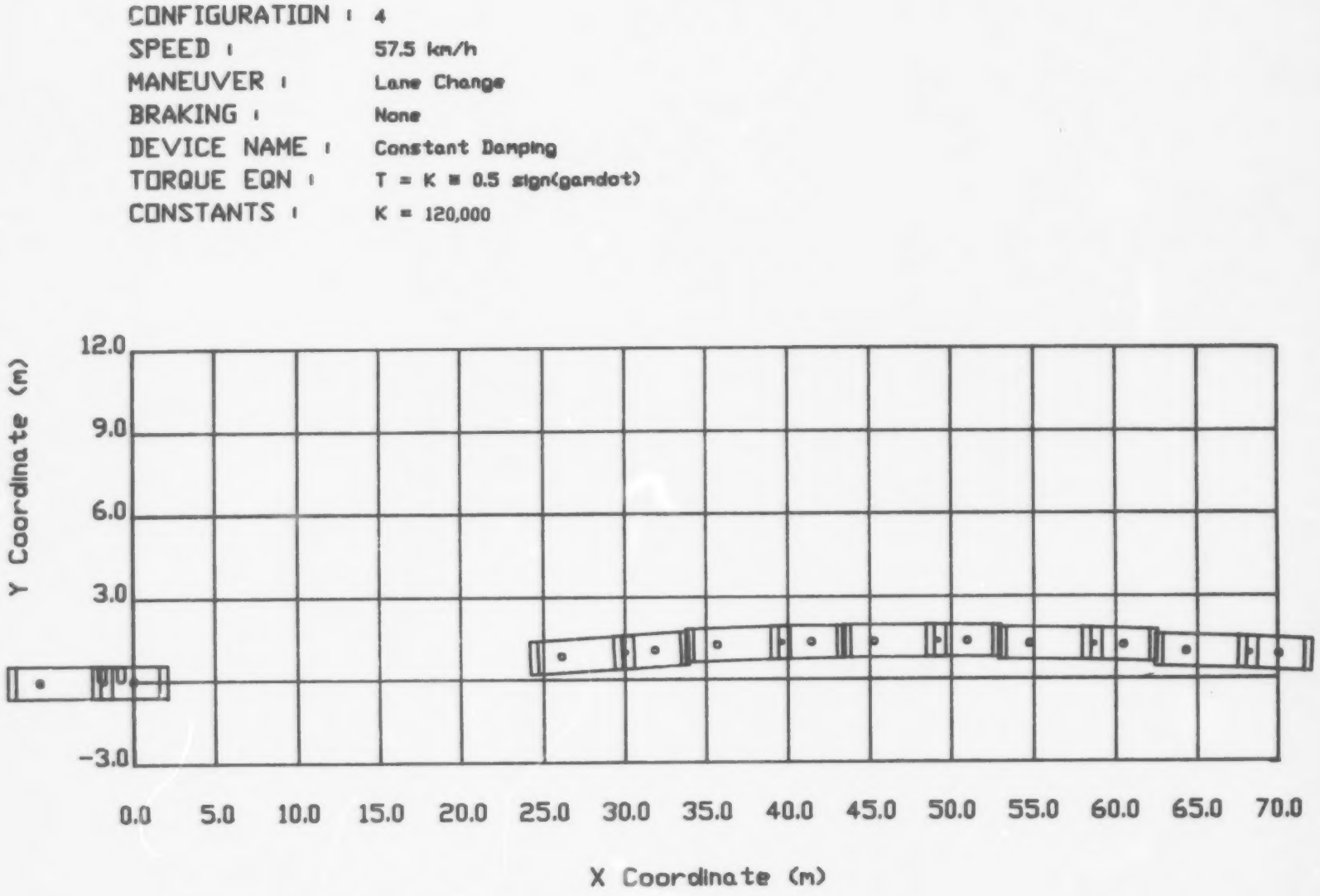


Figure 40 Constant damping Device used in an Unbraked Lane Change Manoeuvre

Figure 41 Constant Damping Device (N = 80,000 N) used on Configuration 1

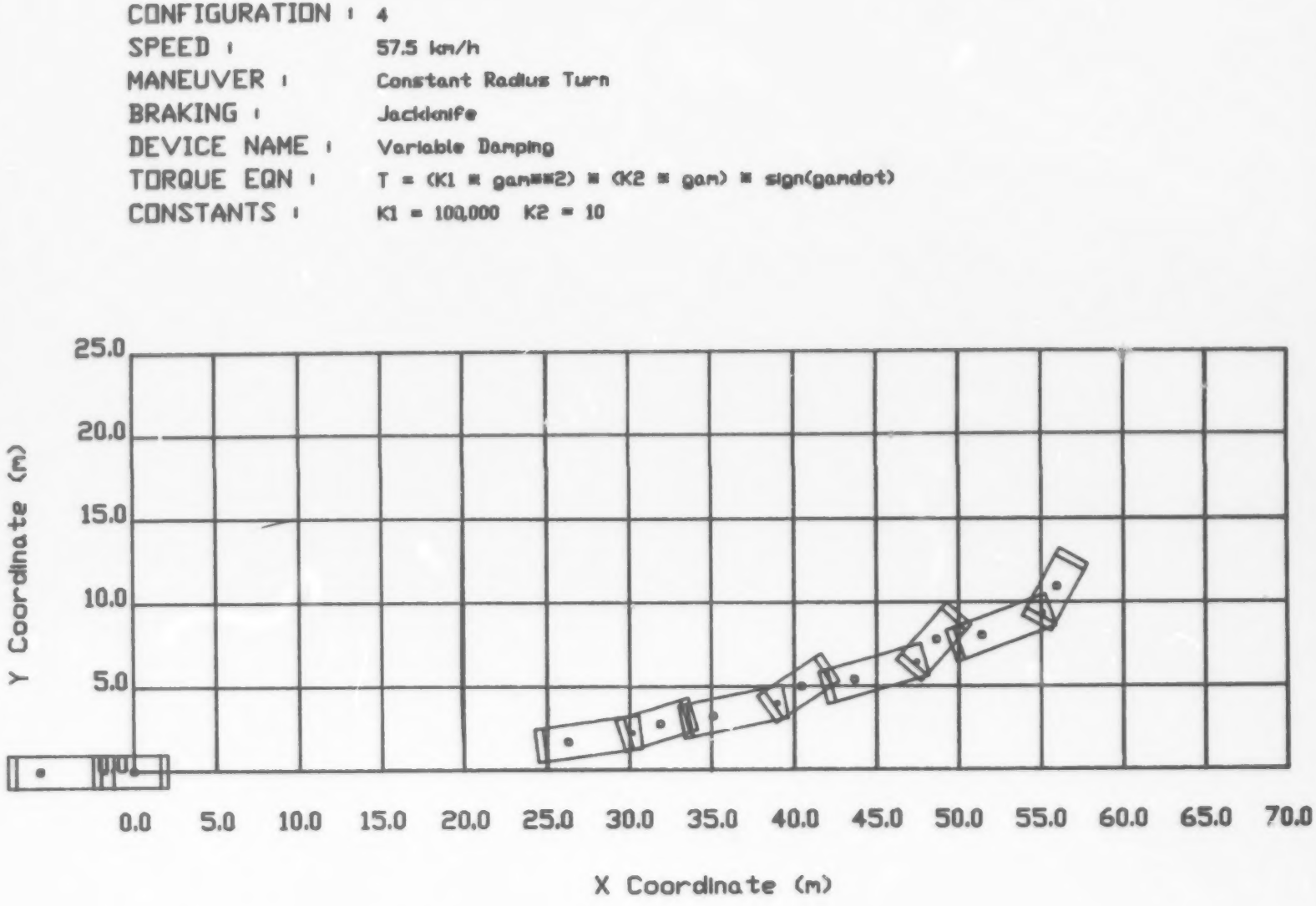

CONFIGURATION : 4
SPEED : 57.5 km/h
MANEUVER : Constant Radius Turn
BRAKING : Jackknife
DEVICE NAME : Variable Damping
TORQUE EQN : T = (K1 * gam**2) * (K2 * gam) * sign(gamdot)
CONSTANTS : K1 = 100,000 K2 = 10
25.0
20.0
15.0
10.0
5.0
0.0
Y Coordinate (m)
0.0 5.0 10.0 15.0 20.0 25.0 30.0 35.0 40.0 45.0 50.0 55.0 60.0 65.0 70.0
X Coordinate (m)

Figure 42 Variable Damping Device (K1 = 100,000, K2 = 10)

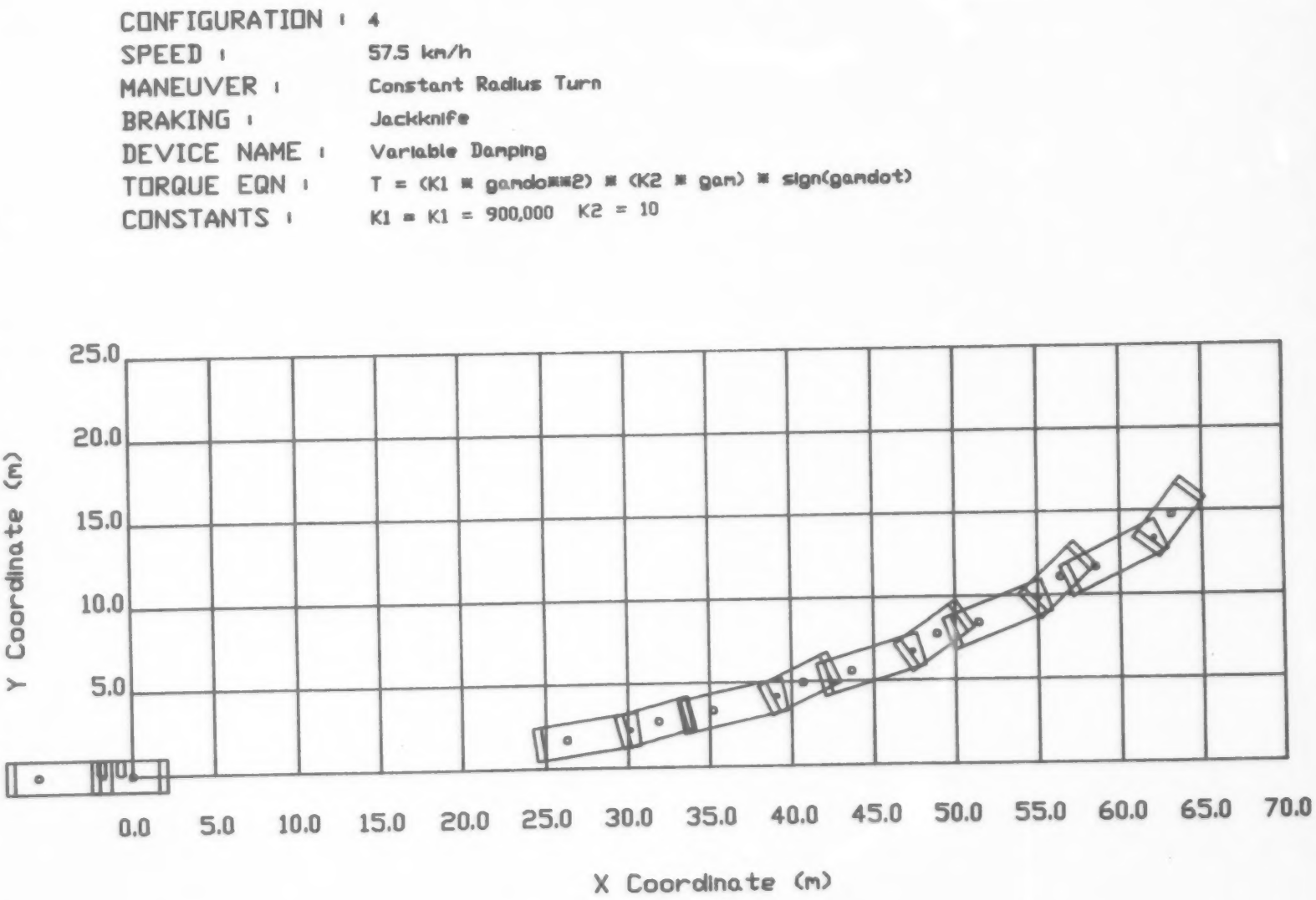
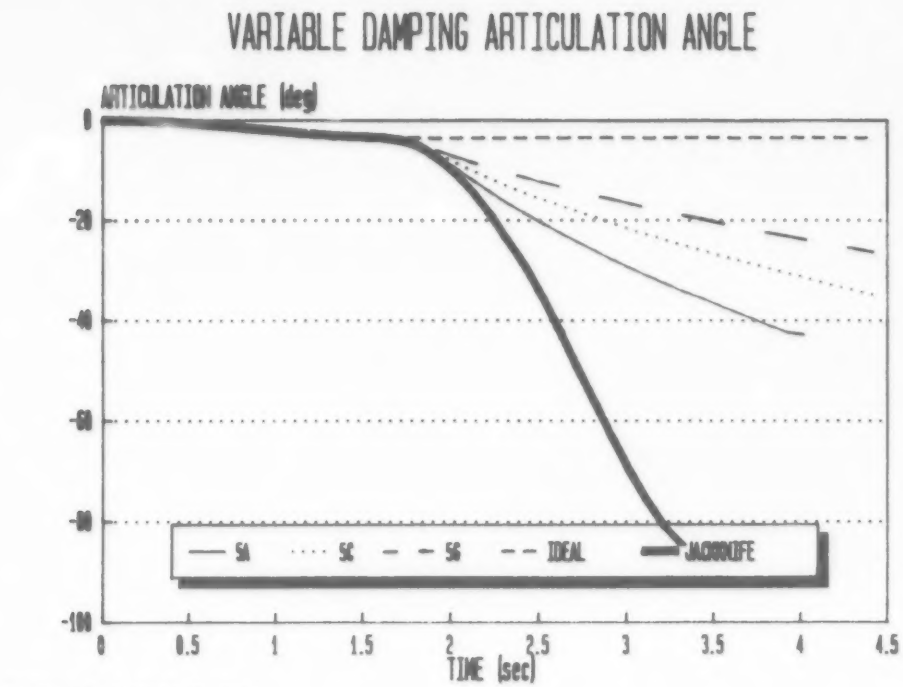


Figure 43 Variable Damping Device (K1 = 900,000, K2 = 10)



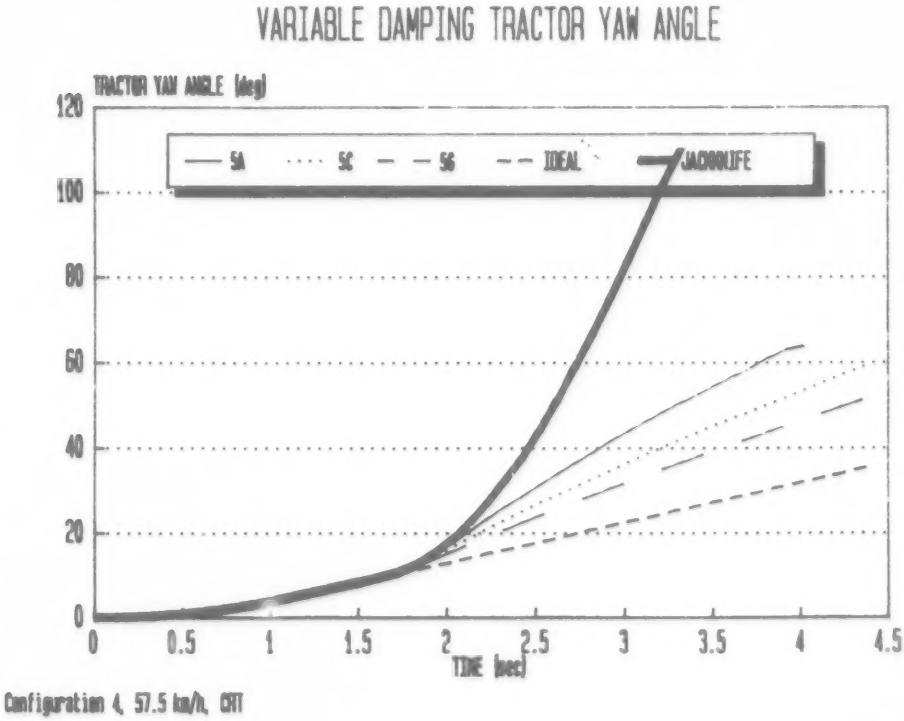


Figure 44 Jackknife Characteristics with a Variable Damping Device

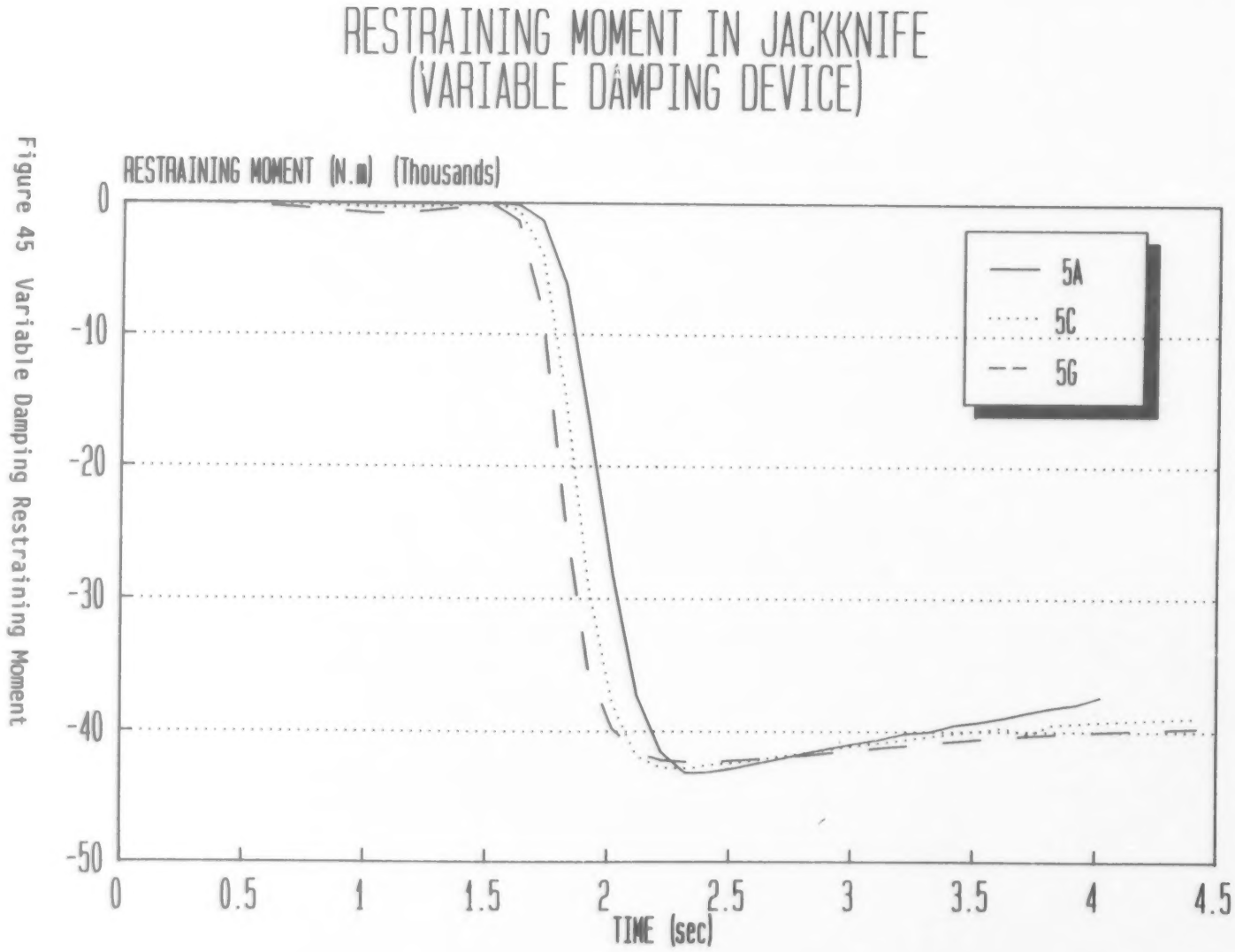


Figure 45 Variable Damping Restraining Moment

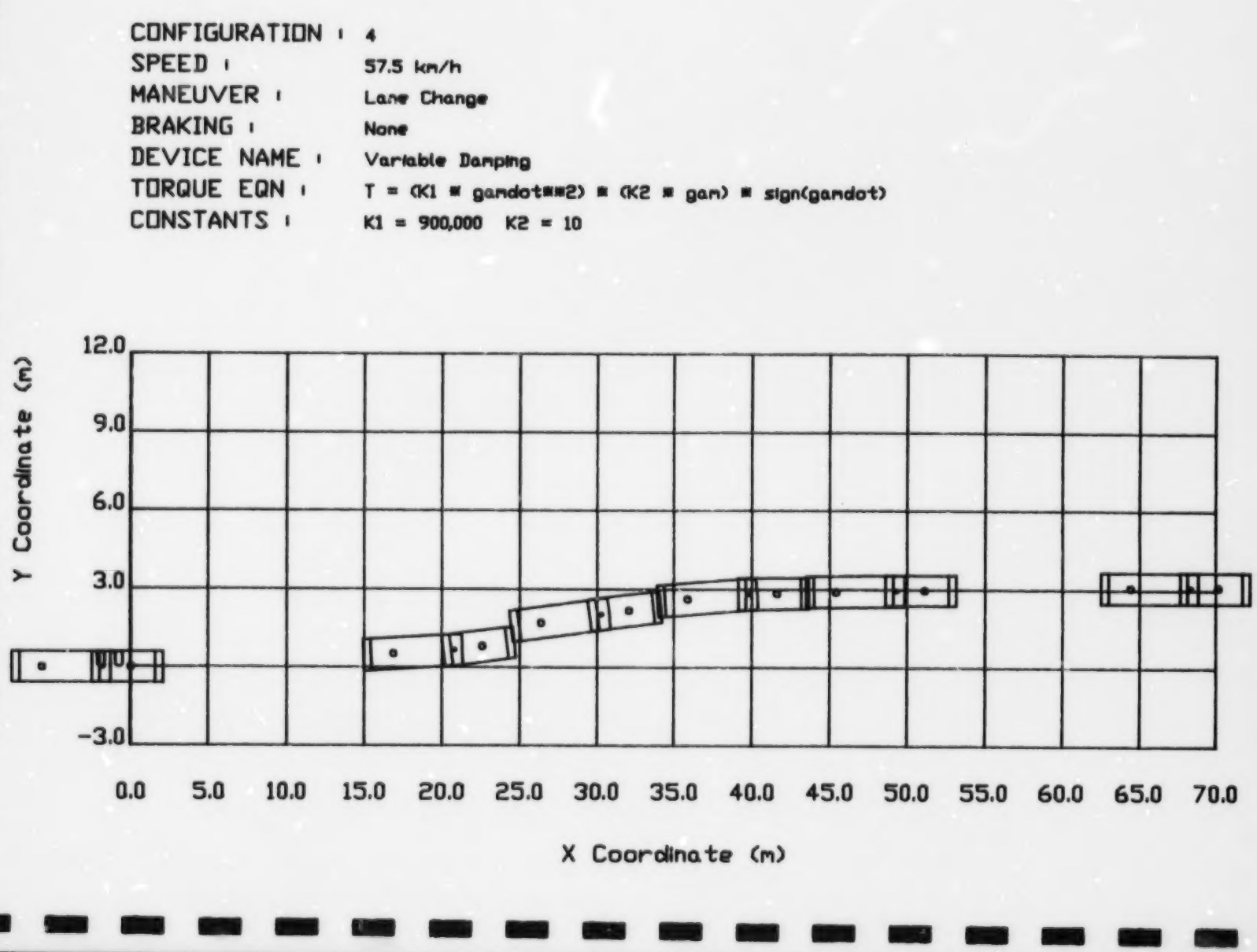


Figure 46 Variable Damping Device Used in an Unbraked Lane Change

Figure 47 Variable Damping Device (K1 = 900,000, K2 = 10) used on Configuration 1

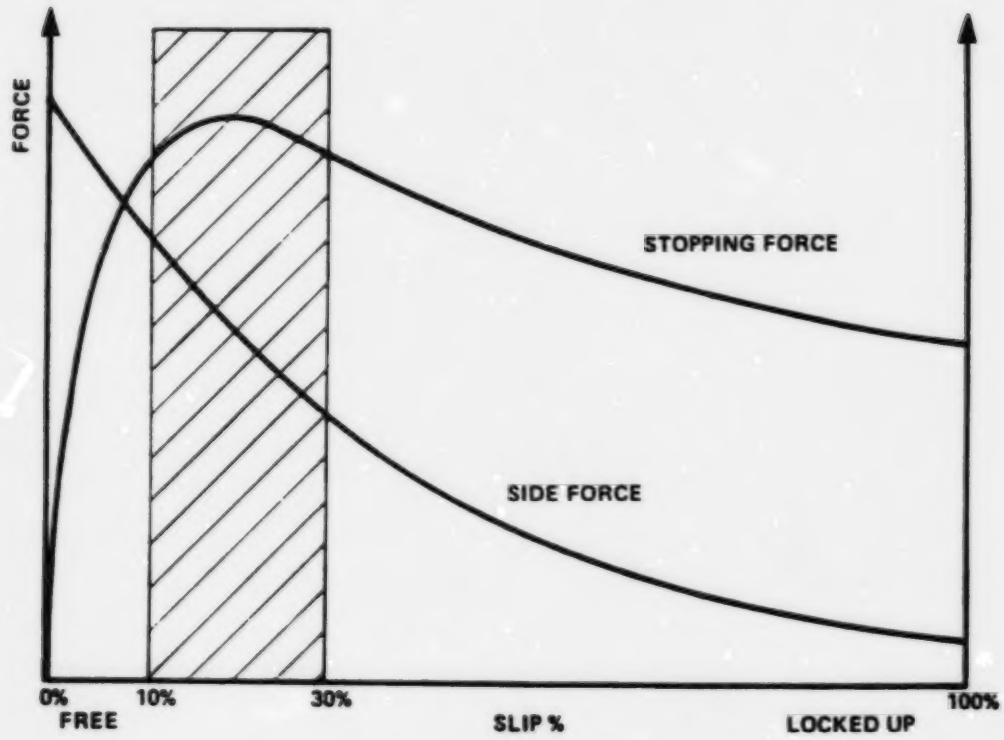


Figure 48 Anti-Lock Brake System Theory

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APPENDIX "A"

Baseline Simulations

Appendix "A" - Baseline Simulations

This appendix includes plots of the baseline manoeuvres of both configurations that were simulated in stable and jackknife situations for the constant radius turn. A series of plots has also been included to show configuration 4 in a stable lane change manoeuvre, and in four different jackknife situations that occur at four different points in time (as discussed in Section 2.2 of the main body of the report).

In all simulations jackknife was induced through rear tractor tire lock-up, and the peak and sliding friction values (μ_p and μ_s) remained constant at 0.47 and 0.25 respectively. The following is a list of the figures included:

Figure		Page
Figure A1	Configuration 1 in an Unstable Constant Radius Turn Manoeuvre	A2
Figure A2	Configuration 1 in a Brake Induced Jackknife on a Constant Radius Turn	A3
Figure A3	Configuration 4 in an Unstable Constant Radius Turn Manoeuvre	A4
Figure A4	Configuration 1 in a Brake Induced Jackknife on a Constant Radius Turn	A5
Figure A5	Configuration 4 in an Unbraked Lane Change Manoeuvre	A6
Figure A6	Configuration 4 Braking Before a Lane Change (J1)	A7
Figure A7	Configuration 4 Braking at the +5° Steer Input Peak (J2)	A8
Figure A8	Configuration 4 Braking at the Positive to Negative Steer Input Transition Point (J3)	A9
Figure A9	Configuration 4 Braking at the -5° Steer Input Peak (J4)	A10

CONFIGURATION : 1
SPEED : 57.5 km/h
MANEUVER : Constant Radius Turn
BRAKING : None
DEVICE NAME : No Device
TORQUE EQN : -
CONSTANTS : -

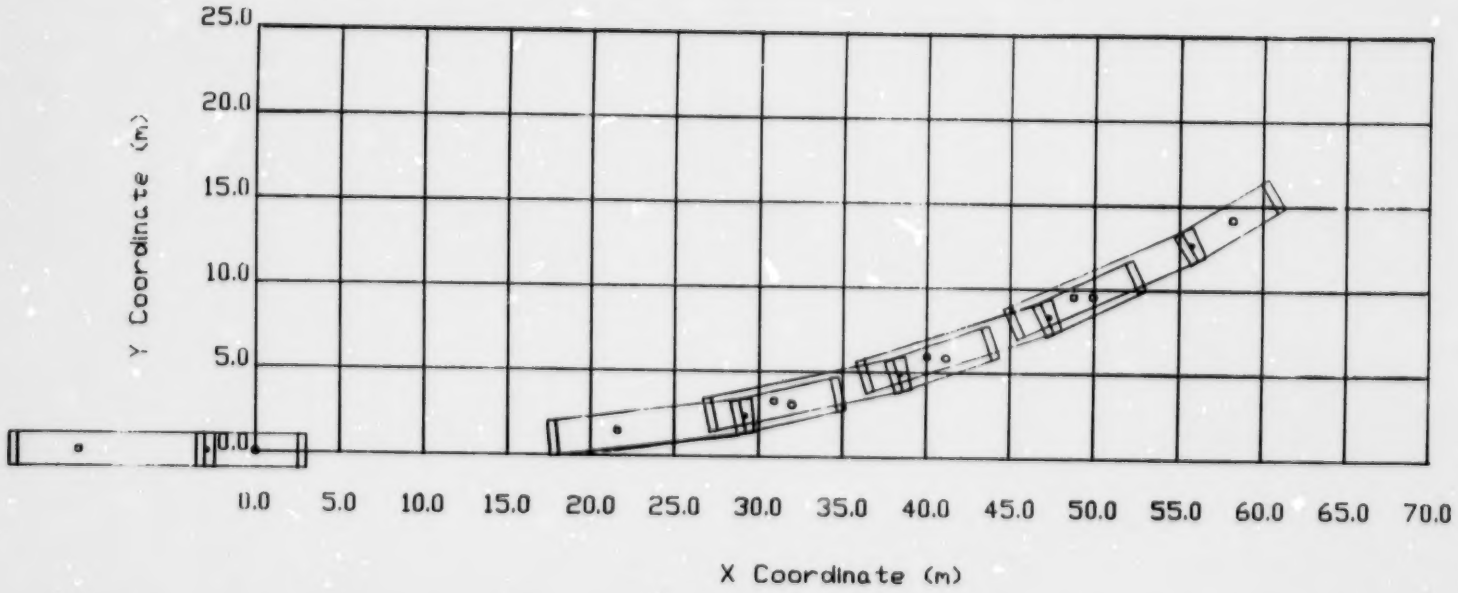

Figure A1 Configuration 1 in an Unstable Constant Radius Turn Manoeuvre

Figure A2 Configuration 1 in a Brake Induced Jackknife on a Constant Radius Turn

CONFIGURATION : 4
SPEED : 57.5 km/h
MANEUVER : Constant Radius Turn
BRAKING : None
DEVICE NAME : No Device
TORQUE EQN : -
CONSTANTS : -

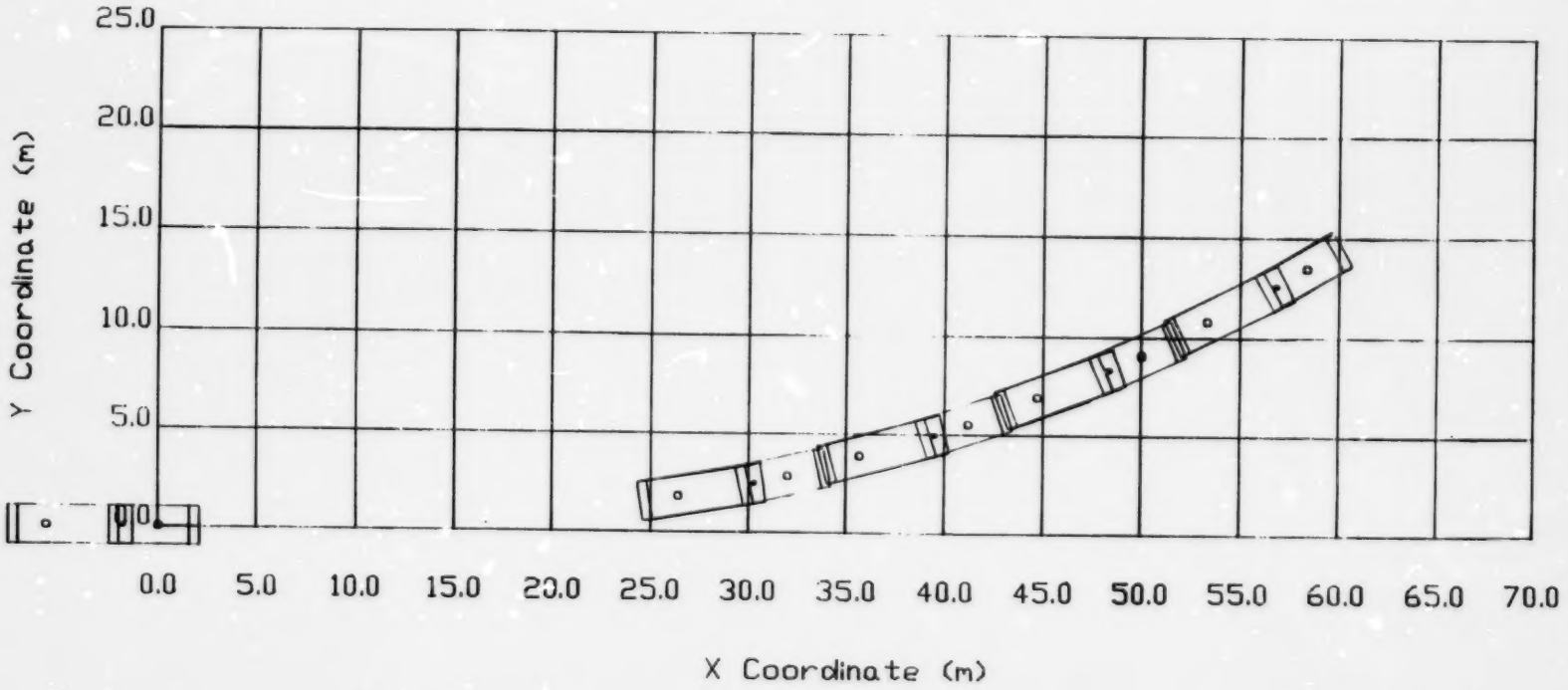

Figure A3 Configuration 4 in an Unstable Constant Radius Turn Manoeuvre

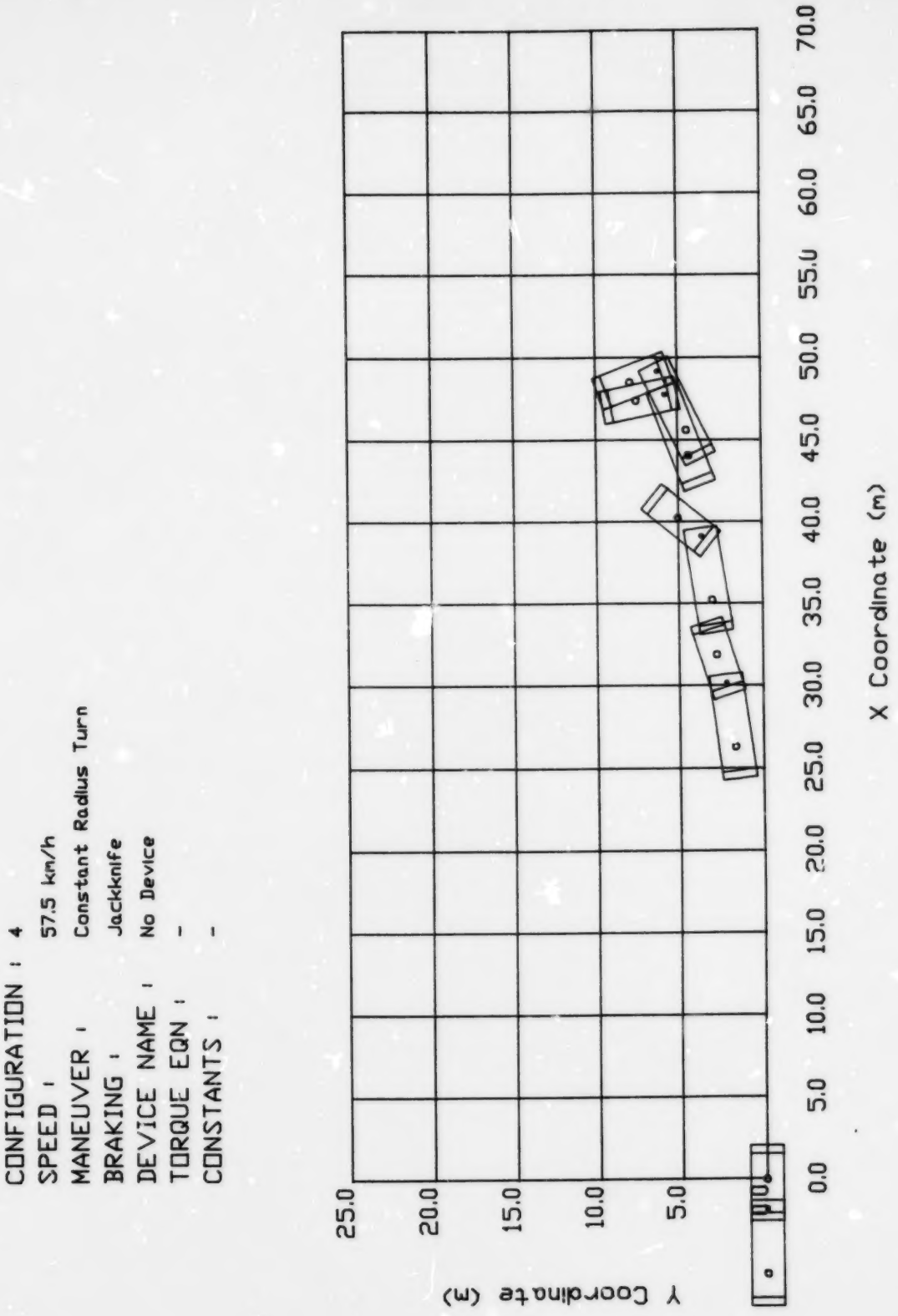

Figure A4 Configuration 1 in a Brake Induced Jackknife on a Constant Radius Turn

Figure A5 Configuration 4 in an Unbraked Lane Change Manoeuvre

CONFIGURATION : 4
SPEED : 57.5 km/h
MANEUVER : Lane Change
BRAKING : Jackknife
DEVICE NAME : No Device
TORQUE EQN : -
CONSTANTS : -

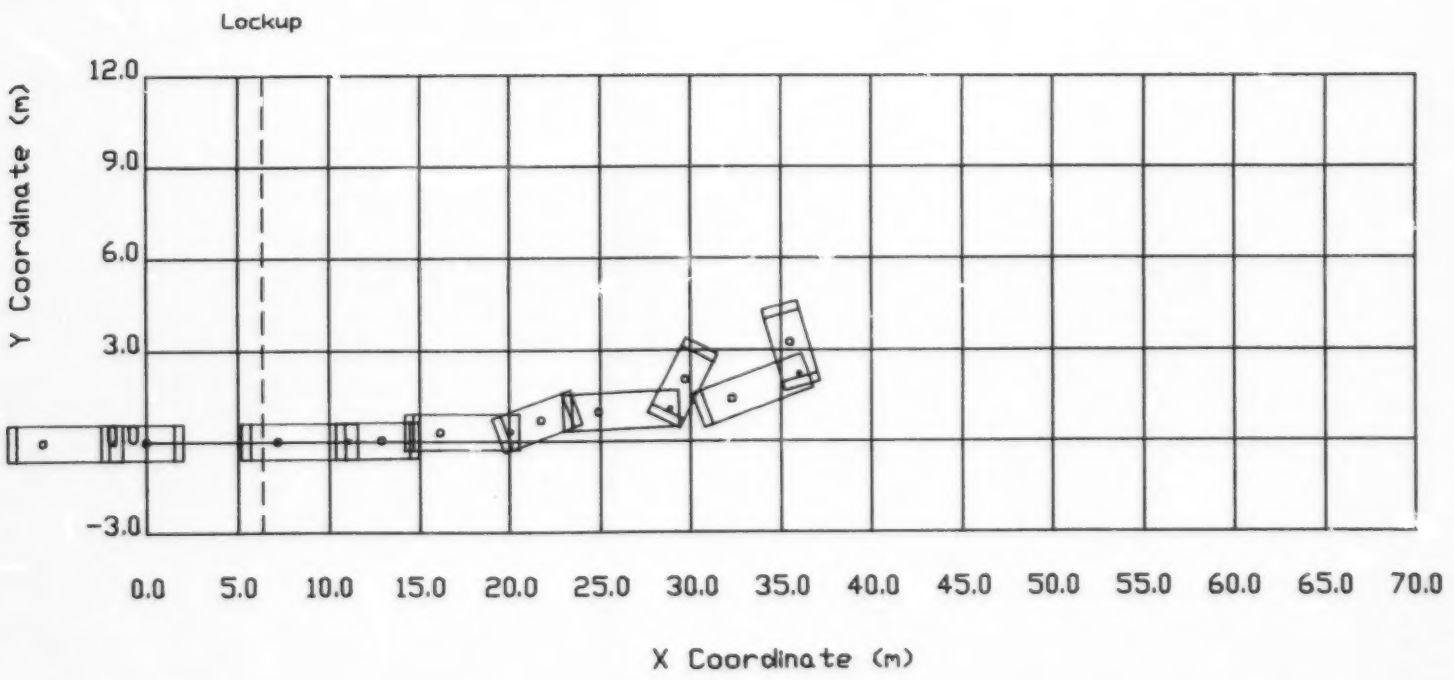

Figure A6 Configuration 4 Braking Before a Lane Change (J1)

Figure A7 Configuration 4 Braking at the +5° Steer Input Peak (J2)

CONFIGURATION : 4
SPEED : 57.5 km/h
MANEUVER : Lane Change
BRAKING : Jackknife
DEVICE NAME : No Device
TORQUE EQN : -
CONSTANTS : -

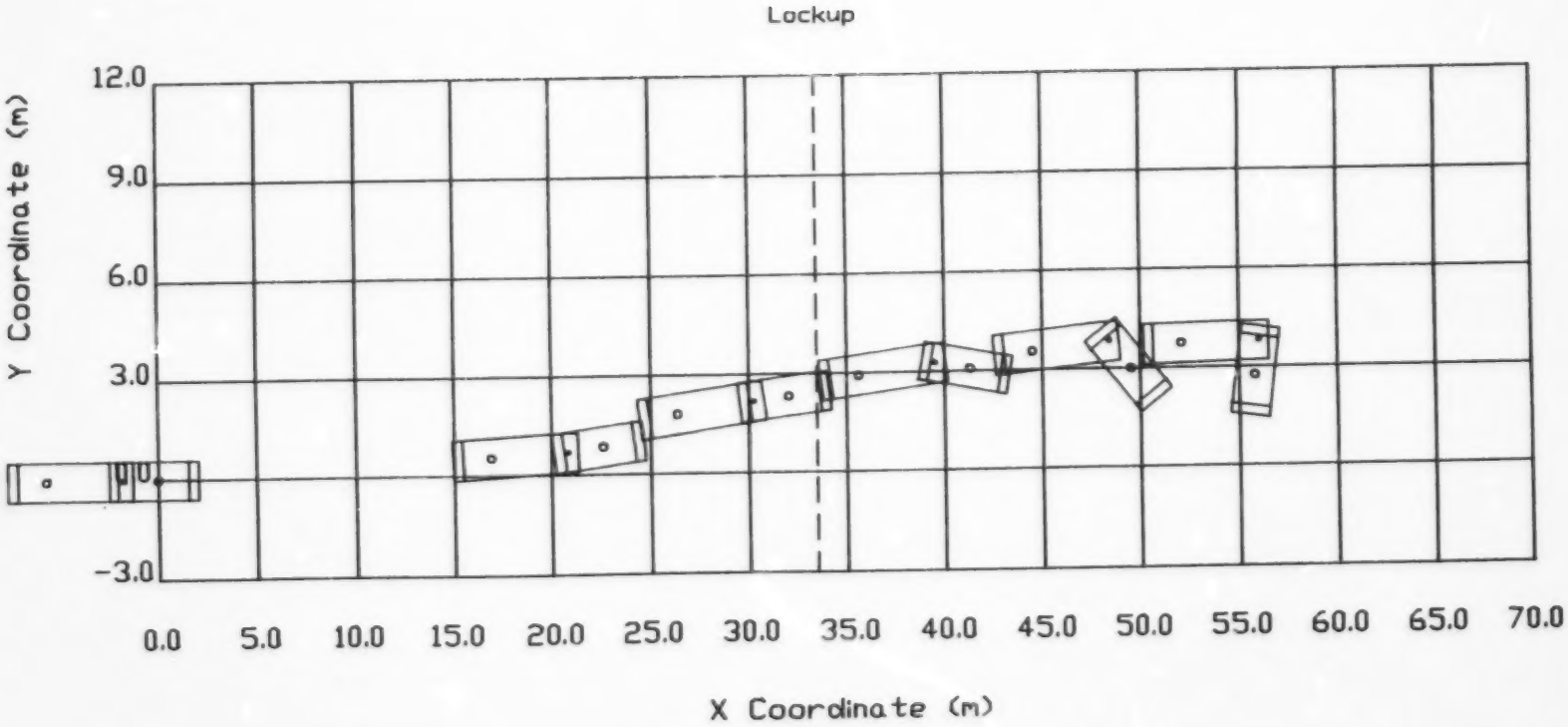

Figure A8 Configuration 4 Braking at the Positive to Negative Steer Input Transition Point (J3)

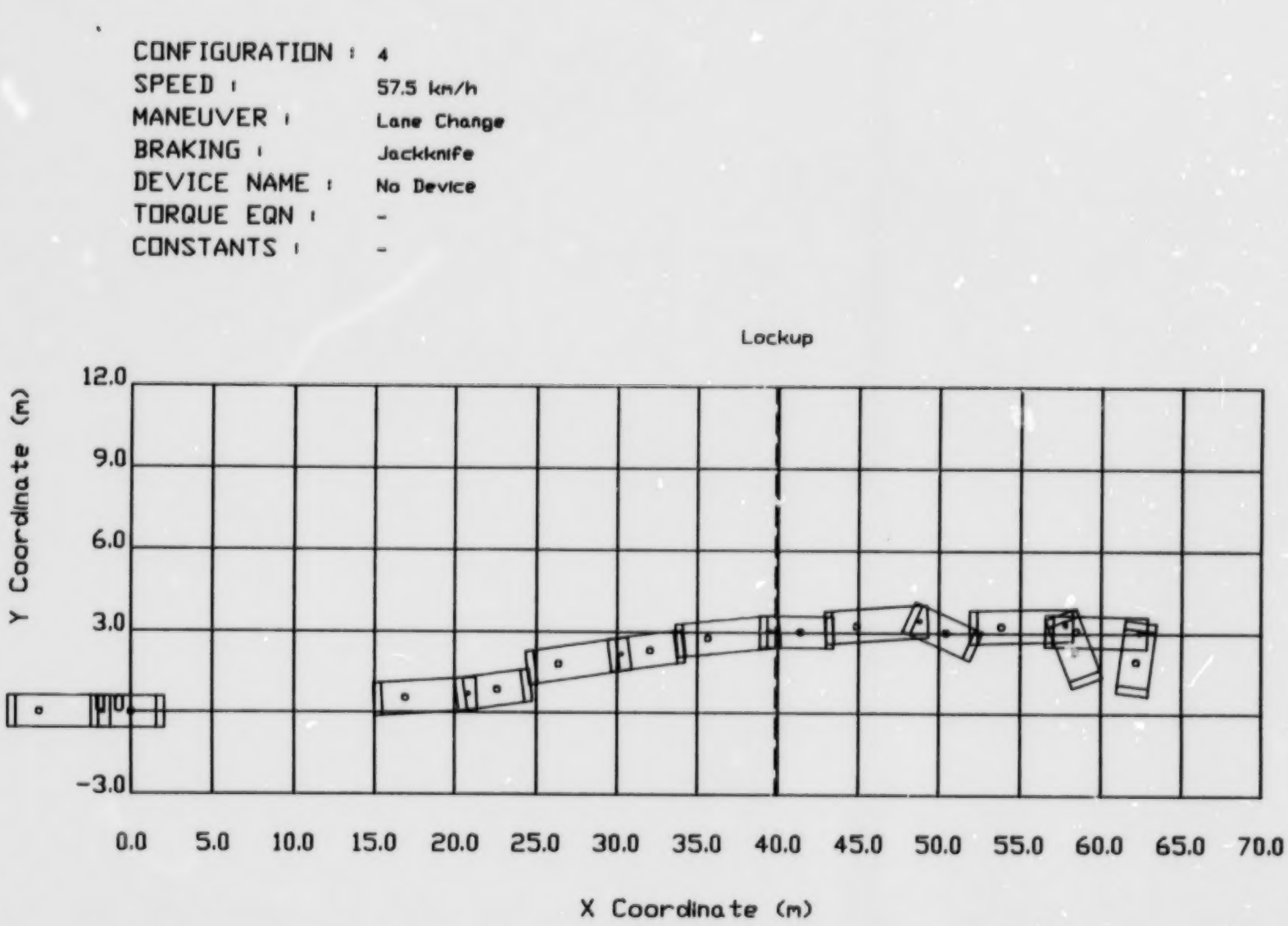

Figure A9 Configuration 4 Braking at the -5° Steer Input Peak (J4)